A TEXT BOOK

OF

NAVAL ARCHITECTURE

FOR THE USE OF

OFFICERS OF THE ROYAL NAVY.

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NAVAL ARCHITECTURE.

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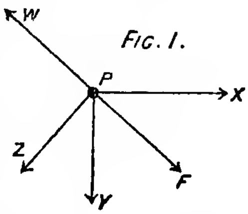
CHAPTER I.

Buoyancy of Ships.

By *buoyancy* is meant the upward support given to a ship by the water in which she is immersed.

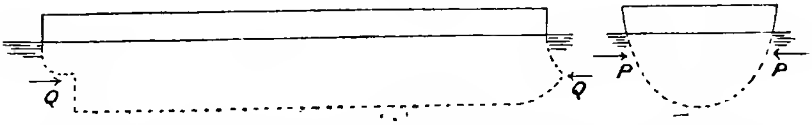
The submerged part of a vessel at rest in still water is subjected to fluid pressure, which acts, at each point, in a direction perpendicular to the surface of the ship at that point. The pressure on any one part may—by the principle of the *resolution of forces*—be supposed replaced by three others having the same effect upon the ship; one acting vertically, the second horizontally athwartships, and the third horizontally fore-and-aft. To readers unacquainted with the above principle this may be made somewhat clearer by an illustration. Imagine a body, denoted by P in Fig. 1, acted upon by three forces X, Y, and Z, along three strings mutually at right angles (like the edges of a box at one corner); these will tend to make the body move in a certain direction, which movement may be prevented by applying a force (W, say) by means of a string in a direction opposite to that in which the body tends to move. Thus the effect of the three forces X, Y, Z, is equal *and opposite* to that of the force W, and the three first-named forces may therefore be replaced by a single force F equal to W and acting in the opposite direction. Conversely, if the force F operated upon the body P, it might be supposed replaced by the three others X, Y, and Z, which have the same effect.

FIG. 1.



We may therefore conceive the pressure upon each part of the ship replaced by three others, thus obtaining three sets of parallel pressures, one set acting vertically, another horizontally athwartships, and the third fore-and-aft; noticing that some of the pressures in any one set may be acting in the opposite direction to others in the same set. The ship being supposed at rest, there is no motion in an athwartship or fore-and-aft direction, and hence

FIG. 2.



the sets of pressures in those directions must respectively balance amongst themselves; that is, the sum of the pressures, P in Fig. 2, on one side of the ship must be equal to the sum of the pressures on the other side; and the total pressure Q urging the ship ahead must be equal to that tending to make the vessel go astern. The value of P is about 3,500 tons in a large ship, and of Q between 400 and 500 tons.

The *vertical* fluid pressures are the most important, as these give the *buoyancy* which supports the ship. Since any number of parallel forces may be supposed replaced by a single force—or *resultant* as it is termed—acting parallel to them and equal to their sum, it is only necessary to determine the magnitude of this resultant and the position of its line of action, in order to find the effect of all the vertical pressures upon the ship. Before proceeding to do this, one or two definitions will be given.

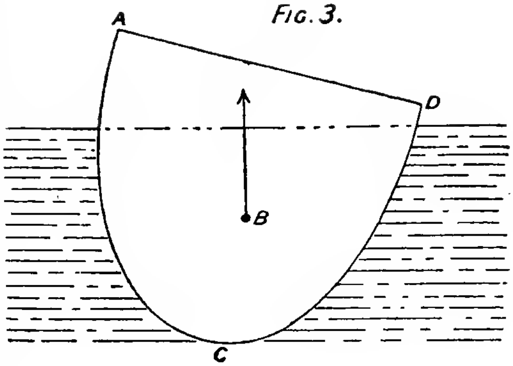
The *centre of gravity* of a body may be defined as the point through which the weight of the body, when at rest, may be supposed to act, in a direction vertically downwards.

When a ship is immersed in water, it occupies a certain space which would otherwise be filled with water; the quantity of fluid so dislodged is called the *displacement*, and may be measured by its volume in cubic feet or its weight in tons; in the former case it is known as the *volume*, and in the latter the *weight* of displacement. When "displacement" alone is spoken of, the *weight* of displacement is usually meant.

The magnitude, and position of the line of action, of the resultant vertical pressure may now be determined by the aid of the following :—

PROPOSITION : When a vessel is immersed in water in any position, it experiences an upward pressure which is equal to the weight of the displacement in that position; and this pressure acts through the centre of gravity of the displacement.

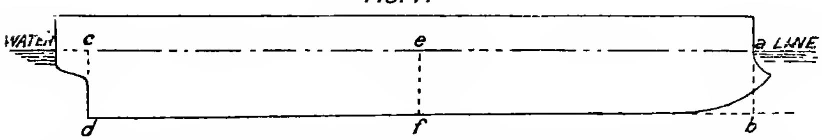
PROOF : In Fig. 3, suppose the vessel A C D held forcibly in the position shown. Imagine the water surrounding this vessel solidified and that then the latter is removed, thus leaving a cavity



having a volume equal to that of the displacement. If now this cavity be filled with water of the same kind as that in which the vessel was immersed, and the solidified water again becomes liquid, there will obviously be no disturbance of level, and the same pressures which were before acting upon the immersed portion of the ship are now acting upon *and supporting* a mass of water having a weight equal to the displacement of the vessel. Now the weight of displacement acts vertically downwards through its centre of gravity, marked B, and therefore, to support this weight, the upward fluid pressure must be equal to the weight of displacement, and its line of action must pass vertically through the point B, since in order that two equal forces may balance each other they must act in the same straight line. Hence also the upward fluid pressure (or buoyancy) on the *ship* must be equal to the weight of water displaced, and must act through the centre of gravity of the displacement; this point is called the *centre of buoyancy*. The situation of this point when the vessel is floating in any assigned position can be calculated and the line of action of the buoyancy thus determined.

When a vessel is upright athwartships, its position in the water is defined by the depth or *draft* of water at the fore and after perpendiculars (between which the length of the ship is measured) to the underside of keel, or that produced; *e.g.*, *ab* in Fig. 4 is

FIG. 4.



the draft of water forward, and *cd* is the draft aft. The *mean* draft of water is that midway between the perpendiculars, as *ef*, and is evidently half the sum of the drafts forward and aft. The difference between the extreme drafts is known as the *trim* of the ship, and when the excess draft is aft, as is usually the case, the vessel is said to trim by the stern. The line in which the surface of the water cuts the surface of the ship when floating in any position is called the *water line* for that position, the area enclosed by that line being the *water plane area.* The line corresponding to the fully laden condition of the ship is the *load water line.*

In the case of a vessel floating freely and at rest in still water, the only forces acting vertically are the weight of the ship downwards through its centre of gravity, and the buoyancy upwards through the centre of buoyancy; and since these balance, the following conditions must hold:—

I. The weight of the ship must be equal to the weight of water displaced.

II. The centre of gravity of the ship and the centre of buoyancy must be in the same vertical line.

A ship being desired which shall have certain weights of armour, guns, &c., an estimate can be made of the weight of the structure (or *hull*) necessary to carry the above safely at sea. The total displacement—being the sum of the weights carried and structure to carry them—can thus be ascertained, as well as the position of the centre of gravity of the ship. Then the second condition of rest mentioned above, teaches that the under-water part of the hull must be so shaped as to bring the centre of buoyancy in a line with the centre of gravity, whilst the first condition enables the *size* of this part of the hull to be determined, since if the *weight* of displacement is known, its *volume* can be calculated when the weight of a cubic foot of the water in which the vessel is to be immersed is known. Sea water, in which war ships principally float, weighs 64 lbs. per cubic foot, so that 35 cubic feet weigh one ton. If therefore the weight of displacement is multiplied by 35, the required volume of displacement in sea water is obtained. Conversely, dividing a known volume of displacement of a ship in sea water by 35 will give its weight. To take a simple example: a box-shaped vessel 210 feet long and 40 feet broad floats in sea water at a draft of 15 feet forward and aft; what is its weight?

The volume of displacement is - $\left.\right\}$ $=210 \times 40 \times 15 = 126{,}000$ cubic feet.

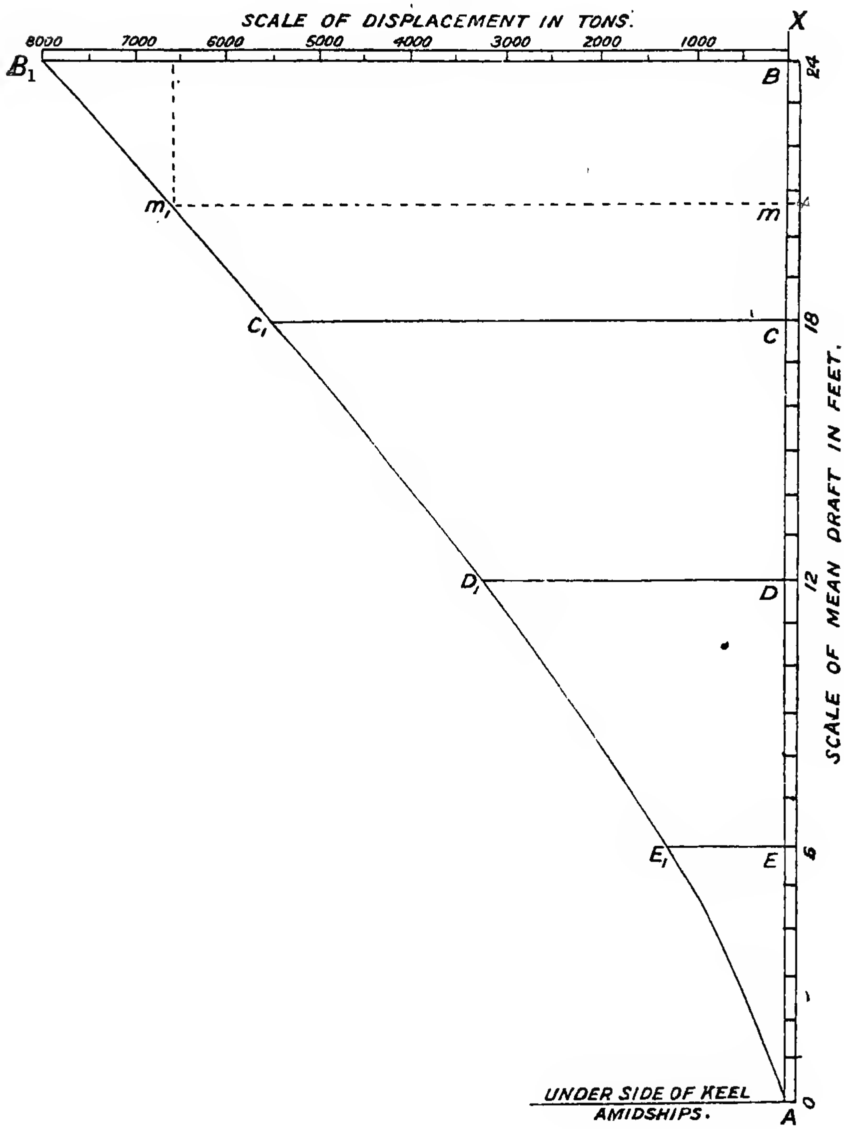
Hence the weight of displacement - - $\left.\right\}$ $=126{,}000 \div 35 = 3{,}600$ tons.

To get the volume of displacement of a ship-shaped vessel when floating at a given water line, the drawings showing her form and size are necessary, the volume being calculated from them by the aid of well-known rules. Thence the *weight* of displacement can be obtained.

Fresh and river water differ in weight from sea water. The former weighs $62\frac{1}{2}$ lbs. per cubic foot, whilst the latter varies in weight according to its position relatively to the mouth of the river; that at the London Docks weighs 63 lbs. per cubic foot. In consequence of this, a ship will sink in the water on passing from the sea into a river, since a greater volume of displacement is necessary in the latter case to make up the same weight of ship. This sinkage is about $3\frac{1}{2}$ inches for a ship of 10,000 tons displacement, and is, conversely, the amount she would rise on passing from a river to the sea.

As the stores, &c., of a ship are consumed, the displacement becomes less, and the ship therefore rises in the water; *e.g.*

FIG. 5.



the *Inflexible* rises 27 inches due to the consumption of her coals alone. It is thus often requisite to know the displacement up to water lines below that for the load condition. To ascertain this without the labour of direct calculation, a "curve of displacement" is used, which is constructed in the following manner:—A vertical line AX, Fig. 5, is taken, on which to set off—usually on a scale of half an inch to a foot—mean drafts, measuring from A, which represents the underside of keel amidships, or zero mean draft. Let AB denote, on the proper scale, the mean draft corresponding to the load displacement. Draw BB_1 perpendicular to AX and let its length represent the load displacement on a certain scale, a quarter of an inch often representing 200 tons displacement for a large ship. The same construction is made for several other water lines parallel to the load water line, the ship being supposed to lighten without change of trim. Let AC, AD, &c., denote the various mean drafts corresponding to these water lines, and CC_1, DD_1, &c., the displacements at those drafts, obtained by direct calculation from the drawings. A curved line drawn through the points, B_1, C_1, D_1, &c., is the "curve of displacement," its use being to ascertain, by a simple measurement, the displacement for any draft between those for which it has been actually calculated. Suppose, for example, that the particular ship of which the curve is given in Fig. 5 floated at a mean draft of $20\frac{3}{4}$ feet on a certain occasion. To ascertain the corresponding displacement at that time it is only necessary to set off Am to represent $20\frac{3}{4}$ feet on the proper scale, and to draw mm_1 perpendicular to AX to meet the curve in m_1. Then mm_1, which represent 6,600 tons on the proper scale, gives the displacement required.

It should be noticed that this supposes the ship to have the same trim at the mean draft of $20\frac{3}{4}$ feet as in the load condition, the curve having been constructed on that assumption; but the displacements obtained from the curve are also very approximately correct for all but extreme departures from that trim.

Ships have a form of displacement curve similar to that shown in Fig. 5, but a particularly simple case is that of a box floating evenly in the water, having the same draft forward and aft. It is evident that here the "curve" becomes a straight line, for at one-half the load draft the displacement is one-half that in the load condition; at one-third the load draft the displacement is one-third the load displacement, and so on, the displacement being proportional to the draft.

The converse case, viz., to determine the draft corresponding to a given displacement can also be easily dealt with; for it is only necessary to draw a straight line parallel to the base line of the curve, at a distance representing the given displacement from it, and the draft corresponding to the point where this line meets the curve is the draft required. In this way also could be found out the sinkage due to putting weights of moderate amount into the ship when floating at any assigned water line; for the draft corresponding to the new displacement can be ascertained as above, and the difference between this and the original water line will be the sinkage sought. This problem can, however, be more easily solved by finding the weight necessary to sink the ship *one inch* from the assigned water line—known as the "tons per inch immersion" at that line—and dividing the given weight by the quantity so found, thus obtaining the consequent sinkage in inches. It is here assumed that the weight necessary to sink the ship one inch from the assigned line is equal to that which will immerse her each additional inch until the total sinkage is reached, an assumption which is very approximately true for moderate changes of draft within the limits at which the ship is likely to be floating on service.

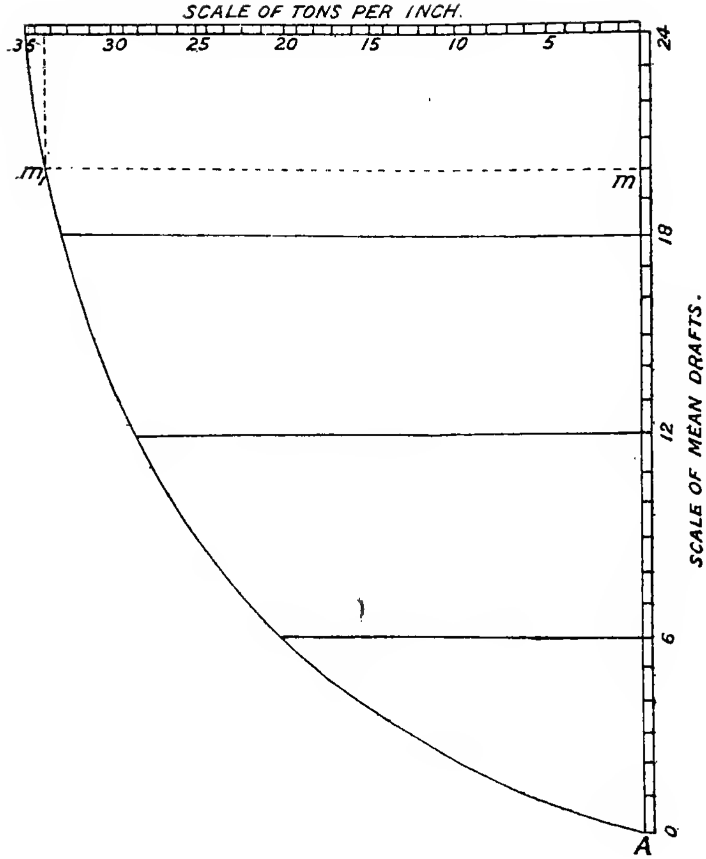
TO FIND THE "TONS PER INCH" AT ANY WATER LINE: The weight which will make the ship sink one inch without change of trim must obviously be equal to the weight of the added displacement, because of the necessary equality between the weight of the ship and the weight of displacement; thus, if A represents the area of the water plane in square feet, the added displacement due to sinking one inch must be $A \times \frac{1}{12}$ cubic feet; *i.e.*, $A \times \frac{1}{12} \div 35 = \frac{A}{420}$ tons.

Hence we get this rule:—

$$\left.\begin{array}{l}\text{Tons per inch}\\ \quad\text{immersion at}\\ \quad\text{any water line}\end{array}\right\} = \frac{\text{Area of water plane at that line in square feet}}{420.}$$

This also gives the weight which must be removed to *lighten* the ship one inch from the same water line. The areas of several water lines parallel to the load line are determined from the drawings, and the "tons per inch" at those lines obtained, the results being set off on a "curve of tons per inch immersion" as shown in Fig. 6. The horizontal measurement at any draft denotes the "tons per inch" at that draft, a quarter of an inch very often representing one ton in the case of a large ship. The use of this curve is that it gives, by a simple measurement, the "tons per

FIG. 6.



inch" at drafts between those for which it has been actually calculated. Suppose, for example, it is desired to know how much the ship having the curve of tons per inch shown in Fig. 6 will sink if 400 tons of coal are put on board when she is floating at a mean draft of 20 feet: set up Am to represent 20 feet on the proper scale, and then mm_1 will give the tons per inch—$33\frac{1}{2}$—at that draft. Hence—

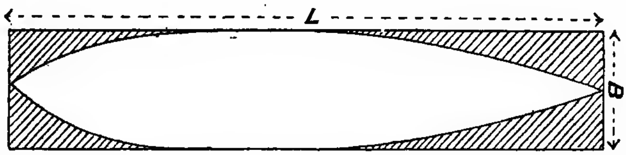
$$\text{Sinkage required} = \frac{400}{33\frac{1}{2}} = 12 \text{ inches very nearly.}$$

An examination of Fig. 6 shows that in the vicinity of the load water line the curve is nearly parallel to the base line, thus justifying the assumption previously mentioned that for moderate changes of draft near the load line the "tons per inch" is practically constant.

Reverting to the box, the displacement of which has been previously found, it is seen that the area of the water plane is the same at all drafts—supposing no change of trim—viz.: 210×40 square feet. Thus the "tons per inch" is $210 \times 40 \div 420 = 20$, and the "curve" of tons per inch is a straight line parallel to the base.

When the drawings of a ship are not available from which to calculate the area of the load water plane, a good approximation to the "tons per inch" for that plane may be made by rules given below; these have been taken by permission of W. H. White, Esq., F.R.S., Assistant Controller of the Navy and Director of Naval Construction, from the "Manual of Naval Architecture." Let L be the length of the ship in feet at the load water line, and B the extreme breadth at that line; then, if the water line were rectangular, its area in square feet would be given by the product $L \times B$, and the "tons per inch" by $L \times B$ divided by 420. The actual area of the water plane is, however, less than that of the rectangle, being

FIG. 7.



"fined" down from the latter at the ends, as shown in Fig. 7, so that the tons per inch for the ship will be obtained by dividing the area of the rectangle by some number greater than 420, its magnitude depending upon the extent to which the plane is "fined" at the ends. If the product $L \times B$ be represented by A, the following rules give, approximately, the tons per inch at the load water line of ships having different degrees of fineness at the ends:—

	Tons per Inch.
1. For ships with fine ends	$= \frac{1}{600} \times$ A.
2. For ships of ordinary form (including probably the great majority of vessels)	$= \frac{1}{560} \times$ A.
3. For ships of great beam in proportion to the length, and ships with bluff ends	$= \frac{1}{500} \times$ A.

In many ships, L is very nearly indeed equal to the length of the ship between the fore and after perpendiculars, in others being a little longer; whilst B is usually equal to the extreme breadth of the ship: in these approximate calculations, therefore, no great error will be involved if the length between perpendiculars and the breadth extreme of the ship are used as the values of L and B respectively.

To illustrate the application of the above rules, suppose the tons per inch at the load line of the *Orlando* to be required; here L=300 feet, and B=56 feet, and as this vessel has fine ends, the first rule will apply; hence

$$\text{Approximate tons per inch} = \frac{300 \times 56}{600} = 28 \text{ tons.}$$

The *actual* tons per inch, as obtained from the drawing, is $27\frac{1}{2}$ tons, so that the approximation is good.

As a ship of ordinary form, the *Invincible* may be taken, having L=280 feet and B=54 feet, so that

$$\text{Approximate tons per inch} = \frac{280 \times 54}{560} = 27 \text{ tons.}$$

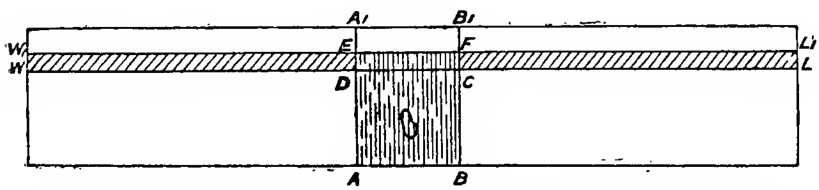
This being also the value as obtained from the drawings.

If any damage occurs to the under water part of a ship, the displacement becomes less than before, due to the entry of water; the vessel therefore begins to sink, and will almost inevitably founder unless structural or other means are provided to limit the accumulation of water. This may be done by employing pumps to eject the water, thus keeping the leak under; or some form of "leak stopper" may be used to prevent the ingress of water, such as the "shot hole stopper mat" for holes near the water line, or "collision mats," sails, etc., in the event of a more dangerous leak below water; or, finally, the interior of the vessel may be divided into numerous watertight compartments by vertical and horizontal partitions, thus restricting the water to the particular compartment in the neighbourhood of the damage. As to the first method, it will be shown (Chapter XI.) that the pumping power supplied to a ship is totally inadequate to the task of keeping down a large leak, such as might occur from the explosion of a torpedo, or the assault of a ram; whilst as to the second, it is extremely difficult to get a leak stopper into position over a jagged hole, through which water is continuously and rapidly flowing. Efficient watertight subdivision must therefore be adopted in all ships, as it must be mainly relied upon to secure the safety of a vessel when greatly damaged below water, the pumps and leak stoppers being regarded as auxiliaries. Details of this subdivision will be found in Chapter X.

If certain compartments of a ship so divided become damaged and filled with water, the buoyancy due to the displacement of those compartments is lost, and the ship will sink until by her extra immersion an additional amount of watertight volume (previously above water) has been put into the water to restore the necessary equality between the weight of the ship and the weight of displacement. The total volume, and corresponding buoyancy, of the watertight space lying *above* the water line is called the reserve of buoyancy, as distinguished from the volume of the ship *below* that line, which measures the buoyancy *utilized.* This reserve of buoyancy usually includes the volume of the ship lying between the water line and upper deck, together with any superstructures, poop, forecastle, etc., which can be closed in and made thoroughly watertight. The preceding has a most important bearing upon the safety of the ship, not only as regards foundering—by being in reserve should a loss of buoyancy occur through damage below water—but also, as will be shown in Chapter II., in respect to her immunity from being capsized. It is usually expressed as a percentage of the displacement, and varies from 10 or 20 per cent. in the low sided American monitors to more than 100 per cent. in high sided vessels with very fine under water forms.

To illustrate the foregoing, and the general method of ascertaining how much a ship will sink when certain compartments are damaged below water, the following example may be taken:— A box-shaped vessel, 280 feet long, 50 feet broad and 27 feet in total depth, floats at a draft of 18 feet forward and aft, and has a central compartment 35 feet long, bounded by two watertight athwartship partitions the whole depth of the box. Find the reserve of buoyancy; and the distance the vessel would sink if the central compartment—supposed empty—were laid open to the sea.

FIG. 8.



In Fig. 8, let AA_1 and BB_1 represent the two partitions, or "bulkheads," and WL the water line at which the box floats before damage. Then, since the volume above water is obviously one-half that below it, the reserve of buoyancy before damage is 50 per cent. of the displacement.

Suppose now that the compartment becomes injured, and that the vessel is forcibly held at the water line WL until the water rises to its own level in the compartment. The part ABCD, which before displaced its own volume of water, no longer does so, and the volume of displacement thereby lost is $35 \times 18 \times 50 =$ 31,500 cubic feet. Thus the box, on being released, must sink bodily in the water without change of trim (the compartment being central) until this lost buoyancy is restored. Let $W_1 L_1$ be the new water line when this is the case, the water consequently rising in the compartment to the height EF. It is thus seen that the part of the *reserve* of buoyancy lying between AA_1 and BB_1 is also lost, since it does not displace water as the vessel sinks; and the ends alone must be looked to in order to supply the lost displacement; that is, the volume W_1EDW, together with the volume FL_1LC must be equal to 31,500 cubic feet. Now the sum of these volumes is obtained by multiplying the intact area of the water plane (*i.e.*, the total area of the plane excluding the part CD) by the sinkage of the ship (WW_1) in feet: this intact area is $(280-35) \times 50 = 12{,}250$ square feet, so that

$$\text{Sinkage of the vessel in feet} = \frac{\text{Loss of buoyancy in cubic feet}}{\text{Area in square feet of intact water plane}}$$

$$= \frac{31{,}500}{12{,}250} = 2\tfrac{4}{7} \text{ feet.}$$

If the compartment had been partially filled with machinery, stores, &c., these would have contributed their own volume of displacement *after* the ship had been damaged, and the loss of buoyancy would have been less than before by an amount equal to the volume of these stores: in other words, the loss of buoyancy is always that due to the *unoccupied space* in the compartment, up to the height of the water line WL. Suppose, for example, that one-fourth the space ABCD had been occupied by stores, the loss of buoyancy, and consequently the sinkage, would only have been three-fourths the amount calculated above. Thus, where convenient, stores such as coal, &c., should be placed in those parts of the ship most liable to damage, as in the region of the water line at the unarmoured ends of central citadel ships (*see* Plate X).

The method used in the example above to ascertain the sinkage is also employed for ship-shaped vessels, the procedure being to first ascertain the unoccupied space in the damaged compartment up to the original water line, and then the area of that part of the water plane which is unaffected by the damage; the division of the former by the latter gives the sinkage required. If the compartment is not central, so that change of trim occurs, the above calculation for the sinkage must be combined with an estimate of the change of trim, in order to get the final position of the ship after sinking.

It has been noticed above that the part of the reserve of buoyancy lying between the athwartship bulkheads was lost when

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the compartment became damaged; this may be avoided by fitting a horizontal watertight partition or "platform" at the height CD, thus preventing the water from rising in the compartment above that height as the vessel sinks; and as the *whole* of the water plane area is now "intact" instead of only seven-eighths as before, the consequent sinkage will only be seven-eighths that previously calculated. Thus the protective watertight decks, worked in many ships approximately at the height of the water line, are very valuable for protecting the *reserve* of buoyancy in the event of damage below water, as well as for preserving the *buoyancy* from the effects of gun fire.

If the platform had been worked, say, 9 feet below water instead of at C D, and the leak occurred below it, the loss of buoyancy would have been only one-half what it was before, since water could not rise above the platform; and as in this case also the total water plane area would be intact, the sinkage would only have been one-half that in the preceding case. If the damage had occurred *above* the platform, the loss of buoyancy would have been the same as when the leak was below it; but as only seven-eighths of the water plane would then be intact, the sinkage would have been greater than in that case, although only one-half as much as when no platform at all is worked in the compartment. It is thus seen that the uses of horizontal watertight platforms below water in case of accident are to lessen the loss of buoyancy, or both the loss of buoyancy and water plane area, according as they are below or above the leak.

Finally, it is seen in the above example that if the athwartship bulkheads were carried to a less height in the ship than E F, water would flow over their tops, as the vessel sank on being damaged, into adjacent compartments—unless the bulkheads were terminated at a watertight platform—and would therefore be of little or no use. Hence the deduction that all the main vertical watertight partitions in a ship should either be carried sufficiently high to ensure that their tops would still be above water after any sinkage, change of trim, or change of heel of the ship which is likely to occur from damage in action; or their tops should stop at a watertight deck or platform.

EXAMPLES.

I. A box-shaped vessel 70 feet long and 20 feet broad has a total depth of 13 feet, and a draft of 10 feet forward and aft in sea water. Express the reserve of buoyancy as a percentage of the displacement. Also, find the displacement in cubic feet and in tons, and the tons per inch immersion.

Answers—30 per cent.; 14,000 cubic feet; 400 tons; $3\frac{1}{3}$ tons per inch.

II. The water lines of a vessel are 4 feet apart, and the displacements up to these lines are respectively 7,800, 6,240, 4,760, 3,360, 2,040, 880, 0 tons, commencing from the load water line. Construct to scale the curve of displacement, and find the displacement corresponding to 18 feet mean draft.

Answer—5,480 tons.

III. The tons per inch immersion at the water lines mentioned in the preceding question are 32·6, 32·3, 31·4, 30·1, 27·7, 23·0, 0 tons respectively. Construct to scale the curve of tons per inch immersion, and find the tons per inch at 18 feet mean draft.

Answer—31·9 tons.

IV. The vessel in question I. has a central compartment 10 feet long, bounded by watertight bulkheads which extend from top to bottom. If this compartment is laid open to the sea, how far will the vessel sink?

Answer—20 inches.

CHAPTER II.

STABILITY OF SHIPS.

A SHIP'S stability is that quality by virtue of which she tends to right herself when inclined from her position of rest.

In the preceding chapter it was shown that when the displacement of a ship is equal to its weight, and the centres of gravity and buoyancy are in the same vertical line, the vessel will be at rest (or in *equilibrium*) in perfectly still water, if subject to no external disturbing causes. In practice, however, this freedom from disturbance cannot be secured, so it becomes important to investigate the behaviour of the vessel when displaced from her position of rest by external forces such as the wind; in other words the *character* of the equilibrium is required. If a ship when *slightly* inclined in any particular direction from her position of rest returns towards that position when the inclining forces are removed, it is said to be in *stable* equilibrium for the given direction of inclination; if on being released it moves farther from its position of rest it is in *unstable* equilibrium, whilst if it remains in the slightly displaced position, without any tendency to return towards, or move farther from, its original position of rest, it is in *neutral* or *indifferent* equilibrium for the given direction of inclination. As an example of a body in stable equilibrium, reference may be made to a pendulum with the bob vertically beneath the point of suspension; the same pendulum with the bob balanced vertically above the point of suspension is in unstable equilibrium, and a uniformly heavy sphere on a horizontal surface furnishes an illustration of a body in neutral equilibrium.

In proceeding to discover the character of the equilibrium of a ship, it may be inclined in either a transverse or longitudinal direction, or in directions between those extremes; but since the former is by far the most important, it is to inclinations in a transverse direction that attention will be chiefly directed, the ship being supposed to maintain the same volume of displacement throughout the inclination.

FIG. 9.

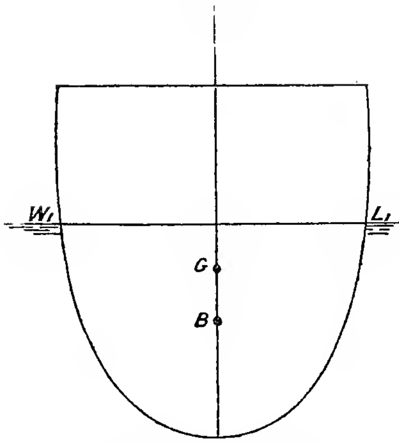
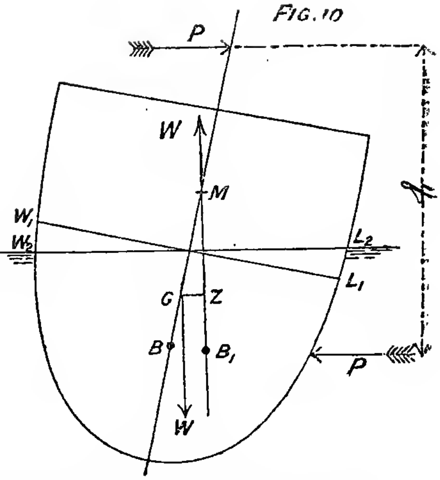


FIG. 10



Let Fig. 9 represent a ship floating freely and at rest in still water at the water line W_1L_1,G marking the height of the centre of gravity and B the position of the centre of buoyancy, G and B being in the same vertical line. Suppose now the vessel is slightly displaced transversely (through 1° or 2°, say) and forcibly held in the position shown in Fig. 10, W_2 L_2 being the new water line. If no weights on board have shifted during the inclination, the centre of gravity is in the same position in the ship as before, but since the *shape* of the volume of displacement has altered, the centre of buoyancy will now be at some point B_1 out of the middle line of the ship; and the line of action of the buoyancy will be that drawn vertically upwards through B_1. This line of action may, or may not, pass through G. If it does, the two conditions of equilibrium are satisfied *after* the ship is inclined, and she will therefore remain at rest in that position; that is, the original position of rest is one of *neutral* equilibrium. Generally, however, the line of action of the buoyancy will not pass through G, but will cut the originally vertical line B G in some point M either above or below G. Suppose it to cut above G, as in Fig. 10; then the weight of the ship (W tons, say) acting downwards through G, and the buoyancy upwards through B_1, form two equal and opposite forces acting on the ship; that is they constitute a "mechanical couple," the effect of which would be to turn the vessel towards her upright position if the external forces were removed which are forcibly inclining the ship. Thus when M is above G, the ship is in *stable* equilibrium. On the other hand, if M fell below G, the weight and buoyancy would form an upsetting couple, and the ship would be in *unstable* equilibrium when upright. It is therefore seen that the character of the equilibrium depends entirely upon the position of G relatively to the point M. This latter point is called the *metacentre*, probably because in a given vessel it is the "meta" (or *limit*) beyond which the centre of gravity may not rise for stable equilibrium. All ships should be stable when ready for sea, and hence the designers must adopt means to ensure that G falls below M in that condition. Now the vertical distribution of the weights of a ship, such as guns, armour, &c., determines the height of the centre of gravity, and since this distribution is practically fixed by given conditions in the design, such as the height at which guns and armour should be carried at sea, little control can be exercised over the vertical position of the centre of gravity. By altering the shape of the ship at the load line, however, and varying the under-water form, the metacentre can be made to occupy very different positions in the vessel whilst maintaining the same volume of displacement; it is high up in broad, shallow ships, and low down in narrow, deep ships. Hence, the naval architect, after calculating the position of the centre of gravity of a new ship, adopts such a form at the load water line, and under water, as will make the metacentre fall above G. The position of the metacentre in a given ship is obtained by calculating, from the drawings, its distance from the centre of buoyancy of the vessel when upright, (which distance for a given volume of displacement is entirely dependent upon the size and shape of the water plane area) the actual position of that centre of buoyancy being also obtained from the same source. The found value of BM being set off from the known position of B will give the point occupied by M in the ship. This marks the point, in the originally vertical line, through which the line of action of the buoyancy passes when the ship is *slightly* inclined from the vertical; but, in ships of usual form, this holds good, approximately, for angles up to 10° or 15°. Hence, when once the position of M has been determined, the line of action of the buoyancy for any given angle less than the above can be obtained by drawing through M a straight line making the given angle with the vertical.

The distance between G and M is called the *transverse metacentric height* or, shortly, the *metacentric height*: its value is very different in different classes of vessels, and has to be adjusted to meet conflicting claims. For example, in order that the ship may be *stiff*—*i.e.*, difficult to incline by external forces such as wind pressure on sails—the metacentric height should be great; whereas to ensure the ship forming a steady gun platform at sea, GM must be made small. The second statement is shown to be true in

Chapter III.; the truth of the first can be demonstrated as follows :— From G in Fig. 10, draw GZ perpendicular to B_1M; GZ is the *arm* or *lever* of the mechanical couple formed by the weight and buoyancy; and as the turning effect of any couple is measured by its *moment* (obtained by multiplying either force into the perpendicular distance between them), the effort which the ship makes to right herself is W × GZ foot tons, W being expressed in tons and the length of the arm GZ in feet. This effort of the ship to right herself when inclined at any angle, is called her *statical stability* at that angle, and must be equal and opposite to the effort of the external forces keeping her inclined; and since to counteract the turning effect of a couple another couple of equal moment must be employed, it follows that the external forces necessary to hold a ship at a given angle must form a couple, the moment of which is equal to the statical stability at that angle. Suppose, for example, the ship is inclined by a wind pressure of P tons acting on the sails; this will cause the ship to have leeway until a corresponding pressure P acts upon her under water surface, after which a uniform motion to leeward will be established, and the vessel will remain at a steady angle of heel. Let the forces P act as shown in Fig. 10, the perpendicular distance between them being h feet. Then

$$P \times h = W \times GZ$$

Now, for the moderate angles of heel usually experienced by ships under wind pressure in smooth water, the line of action of the buoyancy passes through the metacentre, so that GZ = GM sin. θ, where θ is the inclination in degrees. Thus we may write

$$P \times h = W \times GM \sin. \theta$$

This equation enables the angle of heel to be determined, to which a given wind pressure will incline a particular ship: *e.g.*, let W=8000 tons, GM = $3\frac{1}{2}$ feet, P = 14 tons and h = 100 feet; then

$$\text{Sin. } \theta = \frac{14 \times 100}{8000 \times 3\frac{1}{2}} = \frac{1}{20}$$

Whence, from a table of sines, $\theta = 3$ degrees nearly. The above equation also shows that in a ship of given weight, the greater the metacentric height (GM) is, the greater will be the moment of the external forces (P × h) necessary to hold the ship at a given angle θ; or, in other words, the greater the metacentric height is made, the smaller will be the angle to which an external couple of given moment will incline the vessel. Hence, as stated above, to ensure that the ship shall not be easily inclined by external forces, the metacentric height should be made as great as possible.

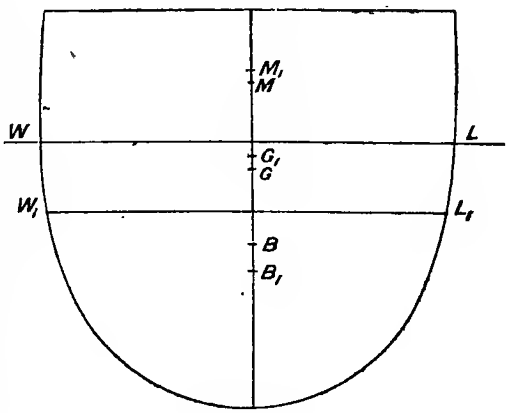
Seeing, then, that for stiffness the metacentric height should be *great*, and for steadiness in a seaway *small*, the GM actually given is mainly determined by experience with successful vessels; such a form of ship being adopted as will make the metacentric height small enough to secure steadiness, without undue sacrifice of stiffnees; observing that the special forms adopted in certain vessels from other considerations—such as in central citadel armour-clads to provide against the reduction of metacentric height which would occur, in the probable event of their ends at the water line being damaged in action; or in armoured coast defence vessels where moderate draft is necessary—often lead to metacentric heights greater than are required from considerations of stiffness alone. The following table shows the values given in various important classes of vessels :—

Name and description of ship.	Metacentric Height.
	Feet
1. *Northumberland:* early broadside armour-clad	$3\frac{1}{3}$
2. *Devastation:* Mastless turret ship	$3\frac{3}{4}$
3. *Inflexible:* do. (central citadel)	$8\frac{1}{4}$
4. *Monarch:* masted turret ship	$2\frac{1}{3}$
5. *Glatton:* coast defence vessel	7
6. *Medina:* gunboat for river service	$12\frac{1}{4}$
7. *Sealark:* sailing brig	$4\frac{1}{4}$
8. *Inconstant:* unarmoured cruiser	$2\frac{3}{4}$
9. Modern protected cruisers, having steadying sails only	About 2
10. Seagoing gun boats and gun vessels	$2\frac{1}{4}$ to 3
11. Torpedo boats 1st class	$1\frac{1}{2}$ to 2
12. Do. 2nd class	About 1

A comparison of the *Sealark* and *Medina* well exhibits the influence of form upon the metacentric height. They have practically the same displacement (380 tons for the former against 363 tons for the latter), but whilst the *Sealark* has a breadth of $29\frac{1}{4}$ feet and a mean draft of 13 feet, the *Medina* is 34 feet broad and has a mean draft of only 6 feet; thus M is much higher in the latter ship (*see* page 21), which conduces to a greater metacentric height.

The above values of GM are those which the vessels have when they are fully laden. But as a ship consumes her stores, coals, &c., not only does she float at lighter drafts, but the positions of the metacentre and centre of gravity become altered. It is therefore important to see

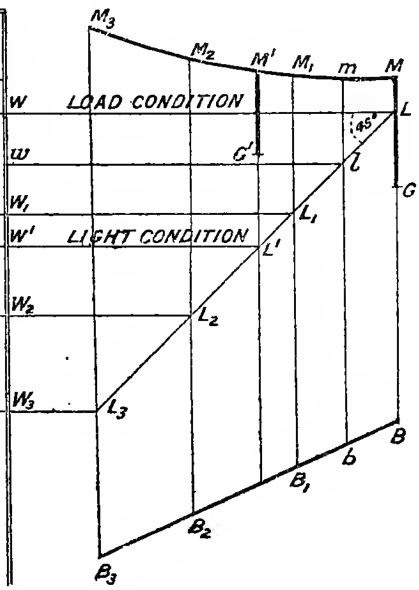
FIG. 11.



that the vessel is sufficiently stiff in the various lightened conditions. Suppose WL is the load-water line of the vessel shown in Fig. 11, B and M being the corresponding centre of buoyancy and metacentre respectively. Also, suppose that after certain stores are consumed the water line is at W_1L_1. The centre of buoyancy (B_1) corresponding to W_1L_1 can be ascertained from the drawings, as also the distance (B_1M_1) between this point and the new metacentre M_1. Thus the position of the metacentre in the lightened condition may be set off, and will not, generally speaking, coincide with M. The different positions of the metacentre for the several lightened conditions are usually shown on a *metacentric diagram*, which is constructed as follows :—

FIG. 12

TONS PER INCH.	DISPLACEMENT IN TONS.	MEAN DRAFT. FT INS
32·5	7600	23 . 6
32·3	6500	20 . 9
32·2	6150	19.10
31·8	5410	18·0
30·9	4360	15·3



A horizontal line WL, Fig. 12, is taken to represent the load water line, and through a convenient point L in it a straight line LL_3 is drawn at an angle of 45 degrees to WL. A series of equidistant lines W_1L_1, W_2L_2, &c., are then drawn parallel to WL at the proper distance apart (usually on a scale of half-an-inch to a foot) to represent successive water lines at which the ship floats as she is lightened, it being supposed that the ship rises bodily without change of trim. Through the points L, L_1, L_2, &c., where the water lines meet the inclined line LL_3, vertical lines are drawn, and on them are set off the distances LB, L_1B_1, L_2B_2, &c., to represent (on the same scale as above) the depths of the respective centres of buoyancy below their corresponding water lines, these distances having been previously calculated from the drawings. The distances BM, $B_1 M_1$, $B_2 M_2$, &c. between each centre of buoyancy and corresponding metacentre are likewise calculated, and set up from the points B, B_1, B_2, &c. A curve passed through the centres of buoyancy is called *the curve of centres of buoyancy*, that through the points M, M_1, M_2, &c. being known as *the curve of metacentres.* It will be noticed that the line $L L_3$ was drawn at an angle of 45 degrees in order to have the vertical lines in the diagram the same distance apart as the horizontal water lines. On the left hand side of the diagram a table is usually arranged, giving particulars of the displacement &c. at each water line.

The use of this diagram is that by its means the positions of the centre of buoyancy and metacentre for any draft between those for which the positions of the centres of buoyancy and metacentres have been actually calculated, can be easily ascertained. Let $w\,l$ be some intermediate water line for which these particulars are required. From l, where $w\,l$ meets $L L_3$, draw a vertical line to meet the curves of metacentres and centres of buoyancy in m and b respectively. Then the metacentre, when the ship is floating at the water line represented by $w\,l$, is at a distance $l\,m$ (on the proper scale) above that line, and the centre of buoyancy is at a distance represented by $l\,b$ below it, presuming of course that the ship has the same trim as when floating at the load water line.

Having a metacentric diagram given for a particular ship, the metacentric height for any lightened condition can be ascertained when the position of the centre of gravity of the vessel in that lightened condition is known. This latter point can be found in the following manner, if (as is usual) the centre of gravity of the vessel in the *load* condition is known, and the weight and position of the stores to be removed to bring the ship from the load to the particular lightened condition are given.

Let G, in Fig. 11, denote the position of the centre of gravity when the ship is fully laden; and let w represent the weight in tons which has to be removed in order that the vessel may float at the water line $W_1 L_1$. Then if the centre of gravity of this weight before removal happens to coincide with G, the centre of gravity of the ship after the weight is removed will still be at G, and the metacentric height in the lightened condition will be $G M_1$. If, however, the centre of gravity of the weight to be removed is h feet *below* G, the centre of gravity of the ship will *rise* from G to G_1 where

$$G\,G_1 = \frac{w \times h}{W - w} \text{ feet,}$$

W being the weight of the ship in tons when fully laden. Then the metacentric height in the lightened condition is G_1M_1. On the other hand, if the centre of gravity of the weight w is h feet *above* G, the centre of gravity of the ship when lightened will fall *below* G a distance given by the above formula.

If stores of total weight w tons are *added* to a ship of weight W at a distance of h feet above or below its original centre of gravity, the centre of gravity of the ship after the stores are put on board will be above or below its original position a distance of $\frac{w \times h}{W+w}$ feet.

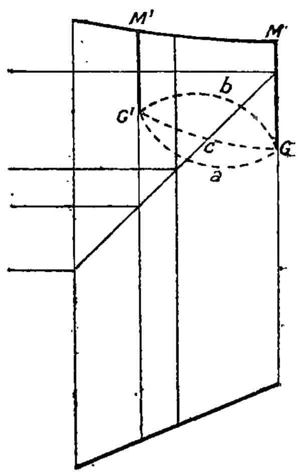
The particular condition to which attention is mainly directed is that when *all* the consumable stores, such as coals, water in boilers, provisions and officers' stores are out of the ship, this being known as the "light" condition. The metacentric height for this condition is shown on the metacentric diagram of a ship, in addition to that for the load condition. To take a numerical example, suppose the weight of consumable stores on board the ship, the metacentric diagram of which is shown in Fig. 12, is 1,450 tons; also that the centre of gravity of this weight is 2·1 feet below the centre of gravity of the ship in the fully laden condition, and that the metacentric height for this condition is 3 feet, so that G M in Fig. 12 represents 3 feet. It is required to find the metacentric height in the "light" condition. When the weights are consumed, since there are 1,100 tons displacement between W L and W_1L_1, the water line for the light condition will be $\frac{350}{32·3}=11$ inches nearly below W_1L_1, the tons per inch at W_1L_1 being 32·3. Let W′L′ represent the water line for this light condition. Then M′ will be the position of the corresponding metacentre. Again, the rise in the centre of gravity due to the consumption of stores is

$$\frac{1450 \times 2·1}{7600-1450}=\frac{3045}{6150}=·5 \text{ feet nearly.}$$

Thus, in setting off the vertical position of G′, the centre of gravity in the light condition, it will be 6 inches nearer to W L than G is. Then G′M′ represents, on the proper scale, 2·6 feet, which is therefore the required metacentric height in the light condition.

The metacentric height of a war ship in the light condition is usually from 6 inches to 1 foot less than its value in the load condition, although in some ships the decrease is greater than this, and in others less. In merchant ships the variation between the load and light conditions is greater than in war ships, and it not unfrequently happens that a vessel which has a good metacentric height in the load condition has a *negative* metacentric height in the extreme light condition. That is to say, in that extreme condition the centre of gravity lies *above* the metacentre, so that the ship is in unstable equilibrium. Such a ship will not remain upright, but will loll over several degrees until it reaches a position of stable equilibrium. Of course this will only occur in harbour, after discharging cargo, and a ship having this characteristic may be rendered stable in the upright position by the introduction of rubble or water ballast low down in the ship, which considerably lowers the centre of gravity without appreciably affecting the position of the metacentre.

FIG. 13.

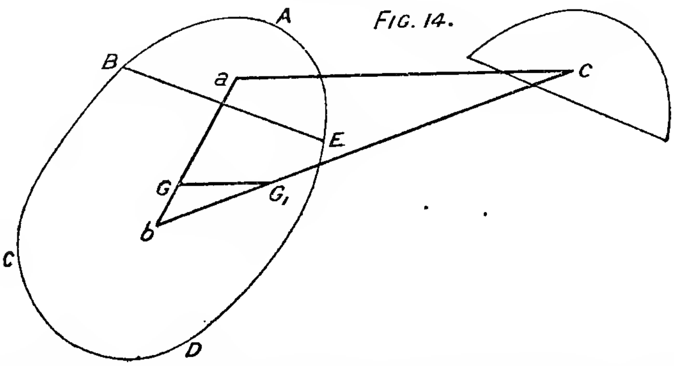


Having the metacentric heights in the extreme light and load conditions, the question arises as to what these heights will be for intermediate conditions of lading. These will depend upon the order in which the stores are consumed. If stores lying high up in the ship are first used, the centre of gravity will first begin to fall, and in going from G in the load to G′ in the light condition, the varying positions of the centre of gravity will be somewhat as represented by the curve G*a*G′ in Fig. 13. In this case the metacentric height is least in the light condition. If, however, the lower stores are first consumed and afterwards any which are very high up, the movement of the centre of gravity will be approximately as shown by the curve G *b* G′, in which case there will be some intermediate condition of lading for which the metacentric height will be less than in either the load or extreme light condition. In most war ships, nearly all the consumable stores are below the centre of gravity for the light condition, so that in whatever order they are consumed the centre of gravity of the ship for intermediate conditions of lading can rise but very little above the centre of gravity for the light condition; hence the successive positions of the centre of gravity will be approximately as shown by the curve G *c* G′. Thus it follows that in the majority of war ships, since the curve M M′ is fairly representative of the curve of metacentres of such ships, it may reasonably be concluded that if sufficient stiffness is secured in the light condition, there will be enough in all other conditions. In those ships, however, which have upper coal bunkers, the movement of the centre of gravity will be of the character shown by G b G′, if the coal and stores are worked out from below before taking coal from those bunkers. This is the more important in ships of moderate dimensions having relatively large stowage in the upper bunkers and a not very great metacentric height (2 ft. say) in the load condition. In such ships it is not advisable to completely empty the lower bunkers before trimming coal from the upper ones.

Not only is the position of the centre of gravity of a ship altered by the consumption or addition of stores, but also by every movement of weights on board. The amount of the change can be estimated by means of the following general mechanical principle: If of a body, of total weight W, a part of weight w is moved through a distance of h feet in any direction, the centre of gravity of the whole will move parallel to that direction through a distance of

$$\frac{w \times h}{W} \text{ feet.}$$

FIG. 14.



Let A B C D E in Fig. 14 represent any body of weight W. Suppose the centre of gravity of the part A B E (of weight w) is at a, and that of the part B E D C at b, the centre of gravity of the whole body being at G. If, now, the part A B E is moved, so that its centre of gravity is taken from a to c, a distance of h feet, the centre of gravity of the whole, after this movement, will be at G_1 on the line joining b and c, where G G_1 is parallel to a c and equal in length to $\frac{w \times h}{W}$ feet. The proof of this depends upon the fact that the common centre of gravity of two bodies lies on the straight line joining their centres of gravity and divides that line in the inverse ratio of the weights of the bodies. Thus b G bears to $b\,a$ the same ratio that the weight of A B E bears to that of ABCDE; *i.e.*, the ratio of w to W. Similarly, $b\,G_1$ has to $b\,c$ the same ratio of w to W. Hence b G is to $b\,a$ as $b\,G_1$ is to $b\,c$, and therefore G G_1 must be parallel to $a\,c$. Again, from the similar triangles b G G_1 and $b\,a\,c$, G G_1 is to $a\,c$ as b G is to $b\,a$, or as w is to W. Expressing this mathematically,

$$\frac{G\,G_1}{a\,c}=\frac{w}{W}.\quad \text{Whence } G\,G_1=\frac{w}{W}\times a\,c=\frac{w\times h}{W}\text{ feet.}$$

Applying the foregoing principle to the case of a ship, it is seen that if any weights are moved *vertically*, the centre of gravity of the whole ship will move in the same direction, and will therefore still remain in the vertical line through the centre of buoyancy. The vessel will thus continue to float at the same water line as before and with the position of the metacentre unchanged, but with an increased or diminished metacentric height according as the weights are moved downward or upward. The lowering or raising of masts and yards is an illustration of this, the consequent alteration in the position of G being easily calculable by the above formula when the weight moved and the distance through which it is transferred are known.

If weights are moved *transversely* across a ship, her centre of gravity will travel out of the vertical line through the centre of buoyancy for the upright position, and the vessel will thus heel over to one side. In Fig. 15 let G and B denote respectively the centres of gravity and buoyancy of a ship when upright, and

FIG. 15.

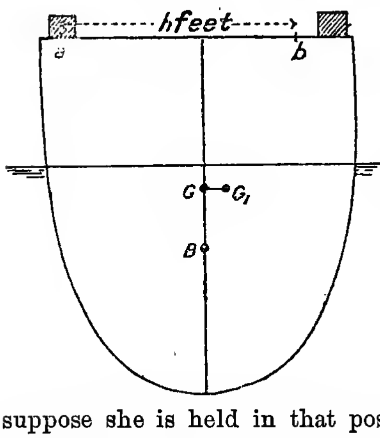
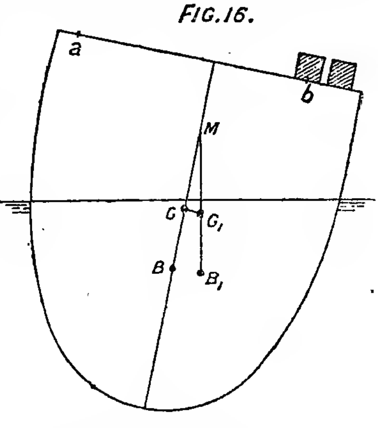


FIG. 16.



suppose she is held in that position whilst a weight of w tons is moved transversely from a to b, a distance of h feet. The centre of gravity of the vessel, after the weight is shifted, will be at G_1 where G G_1 is parallel to $a\,b$ and equal to $\frac{w\times h}{W}$, W being the total weight of the ship. As the buoyancy and weight are now acting in different vertical lines, the ship, on being released from constraint, will incline until the centre of buoyancy is brought vertically beneath G_1 by the altered shape of the displacement. Let Fig. 16 show its position when this is the case, B_1 being the new centre of buoyancy. If the weight moved is moderate in amount, the consequent inclination is small, and the new line of action of the buoyancy will therefore pass through the metacentre, marked M in the Figure. It is now easy to estimate the angle to which a ship of known displacement and metacentric height will be inclined by moving given weights transversely. For from Fig. 16 it is seen that the *tangent* of the required angle is $\frac{G\,G_1}{G\,M}$, so that, calling the angle of inclination θ, and substituting for $G\,G_1$ its value as given above,

$$\tan.\ \theta = \frac{G\,G_1}{G\,M} = \frac{w \times h}{W \times G\,M}.$$

To take an example, suppose that in a ship for which W = 9,000 tons and G M = $3\frac{1}{2}$ feet, a weight of 60 tons is moved transversely through 28 feet. Here

$$\tan.\ \theta = \frac{60 \times 28}{9000 \times 3\frac{1}{2}} = \frac{4}{75} = \cdot053.$$

Referring to a table giving the value of the tangent of each angle, it is found that the angle of which the tangent is ·053 is rather more than 3 degrees, which is therefore the angle required.

On the other hand, if the angle to which the movement of a known weight inclines a ship of given displacement—but the metacentric height of which is not known—is observed, the value of G M may be calculated. For transposing the above equation,

$$G\,M = \frac{w \times h}{W \times \tan.\ \theta} \text{ feet.}$$

For example, moving 60 tons 50 feet across the deck of a ship of 8,000 tons displacement, causes her to incline to an angle the tangent of which is $\frac{1}{16}$ (about $3\frac{1}{2}$ degrees): what is her metacentric height?

$$\text{Here } G\,M = \frac{60 \times 50}{8000 \times \frac{1}{16}} = 6 \text{ feet.}$$

This is the principle of what is known as the *inclining experiment*, which is undertaken when a ship approaches completion, in order to ascertain the exact position of the centre of gravity in that condition. Then, knowing what weights have to go on board, and the weight of ballast &c. to be removed to complete the ship, the centre of gravity in the sea-going condition can be calculated by the aid of principles already mentioned (*see* page 25), and the metacentric height in that condition seen to be sufficient or not. When a ship is designed, a very detailed calculation is made to find out the final position of the centre of gravity; but as during construction several deviations are usually made from the design, it is considered advisable to check the result by a direct experiment. In this experiment, a large number of pigs of iron ballast are arranged on the deck in two equal batches, one on each side of the ship, so that the vessel is still upright when this ballast is on board. The ship is then inclined by moving the ballast on one side to the other, the inclination being measured by means of plumb lines suspended in hatchways or other suitable places.

FIG. 17.

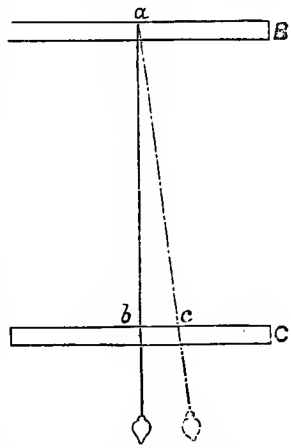
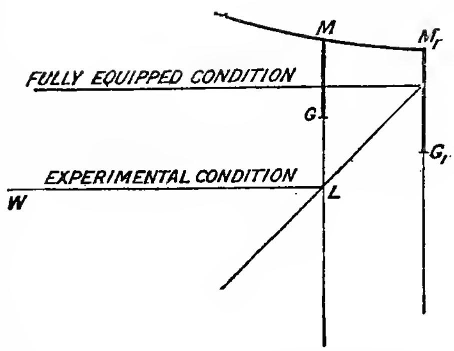


Fig. 17 shows one such line, hung from a batten B, with a second horizontal batten C placed vertically beneath B on which the deviations of the line may be marked. If on moving certain weights from one side of the ship to the other, the plumb line travels from its position $a\,b$ in the upright to $a\,c$, the tangent of the angle to which the vessel is inclined, is evidently $\frac{b\,c}{a\,b}$. The distance $b\,c$ being measured, and that between the battens ($a\,b$) being known, the required tangent can be calculated. Usually three plumb lines are erected in the ship to ensure accuracy; one being forward, the second amidships, and the third aft.

Before the ship is inclined, several precautions have to be taken, the chief of which are to see that there are no weights free to move as the ship inclines, such as water in the bilges or boilers, or hanging weights as rigging, boats, &c.; to select a calm day for the experiment, and to see that the ship is so disposed, as to be affected as little as possible by any light wind that blows. These precautions having been observed, and the ballast and plumb lines being on board, the following is practically the procedure in any particular ship. The drafts forward and aft are carefully noted, and the mean draft thus obtained. Thence the weight (W) of the vessel is known for the experimental condition, from the curve of

FIG. 18.



displacement. Also the position of the metacentre can be found for this condition from the metacentric diagram, assuming that the trim is not very different from its value in the designed load condition. Let Fig. 18 show the metacentric diagram of the ship, W L being the water line at the time of the experiment. Then M marks the height of the metacentre for the experimental condition. The ballast (of weight w) from one side of the ship is now moved to the other through a certain distance (h feet say), and the tangent of the corresponding angle of inclination ($\tan \theta$) is ascertained by means of the plumb lines. Then the metacentric height of the vessel at the time of the experiment is given by

$$G\,M = \frac{w \times h}{W \times \tan. \theta} \text{ feet.}$$

If now this distance is set down below M in Fig 18, the point G so found will be the vertical position of the centre of gravity of the ship in the experimental condition, and thence, knowing the weights to be put on board and taken off to complete the ship, the position G_1 of the centre of gravity in the fully equipped condition can be calculated. In practice, after the deflection due to moving the ballast which is on one side to the other is noted, this ballast is restored to its original position, when the ship should regain the upright if no weights have moved during inclination. The ballast on the *opposite* side is then traversed across the ship through the same distance as the first; and if this gives a slightly different angle from the first experiment, the *mean* of the two results is taken to substitute for *tan.* θ in the above equation.

It should be noticed that if the trim of the ship, when the experiment is performed, is greatly different from that in the designed load condition, independent calculations must be made for the displacement and position of the metacentre, instead of obtaining those quantities from the curve of displacement and metacentres respectively.

A comparison between the positions of the centre of gravity as obtained by calculation and experiment, shows that they usually agree within very narrow limits. For example, in the *Devastation* the difference was $1\frac{1}{2}$ inches, and in the *Alexandra* 1 inch.

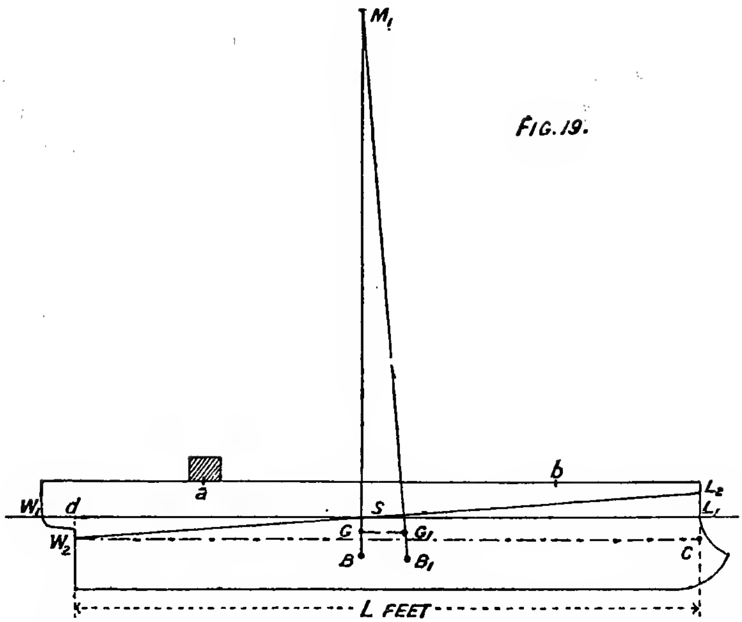


FIG. 19.

Finally, the position of the centre of gravity of a ship is altered by any movement of weights *longitudinally*, causing the vessel to change trim. Suppose a weight of w tons originally at a in Fig. 19 is moved forward to b through a distance of h feet, the total displacement of the ship being W tons. As before, the centre of gravity of the ship will move from its original position G, parallel to $a\,b$, through a distance $G\,G_1 = \frac{w \times h}{W}$ feet. The ship will, in consequence, change trim until the centre of buoyancy moves from its original position B to B_1 vertically beneath G_1. Let $W_2\,L_2$ be the water line when this is the case, the water line before changing trim being shown by $W_1\,L_1$. The vertical through B_1 will intersect the originally vertical line through B in some point M_1 which is called the *longitudinal* metacentre, and the distance between G and M_1 is known as the *longitudinal* metacentric height. This is very much greater than the G M for transverse inclinations, being approximately *equal to the length of the ship*, in a vessel of ordinary proportions; it is rather less than that length in a short ship of deep draft, and considerably more in a long ship of light draft. As a matter of fact the metacentric height of a ship is greatest for longitudinal directions of inclination, and least for transverse directions, and hence the reason for devoting attention mainly to the latter. As the weight w is moved forward, the ship will be depressed forward, and will rise aft so that the same total displacement may be maintained. Hence the two water lines $W_1 L_1$ and $W_2 L_2$ will intersect in a point S, this being the centre of gravity of the water plane $W_1 L_1$ when the inclination is small, and is usually situated a few feet abaft the middle of the length. For all but extreme changes of trim, however, no great error is involved by assuming that the water lines cut at the middle of the ship's length, in which case the rise of the ship aft ($W_2 d$) is equal to the depression ($L_1 L_2$) forward. Of course if the weight had been moved *aft* the depression would have been aft and the rise forward. The trim being the difference between the drafts forward and aft, it follows that the total *change* of trim will be the sum of the increase ($L_1 L_2$) of draft forward, and the decrease ($W_2 d$) aft. Let x feet be the total change of trim, and L feet the length of the ship. Then if $W_2 C$ is drawn parallel to $W_1 L_1$, $L_2 C$ is equal to x, the total change of trim, and $\frac{x}{L}$ is the tangent of the angle to which the ship is inclined. But $\frac{G G_1}{G M_1}$ is also the tangent of the same angle.

$$\text{Hence } \frac{x}{L} = \frac{G G_1}{G M_1} = \frac{w \times h}{W \times G M_1}$$

and therefore the

$$\text{change of trim } (x) \text{ is} = \frac{w \times h \times L}{W \times G M_1} \text{ feet.}$$

$$= \frac{12 \times w \times h \times L}{W \times G M_1} \text{ inches - - (I).}$$

If the longitudinal metacentric height for the particular ship is not known, a fair estimate of the change of trim may be obtained by remembering that $G M_1$ is, roughly speaking, equal to L, and thus

$$\text{approximate change of trim} = \frac{12 \times w \times h}{W} \text{ inches.}$$

The following example will illustrate the use of these rules :—

A ship of 8,000 tons displacement, 300 feet long, and having a longitudinal metacentric height of 315 feet, has a draft of 24 feet forward and 26 feet aft. If now a 45-ton gun is moved aft through 110 feet, what will be the new draft forward and aft?

$$\text{Here change of trim} = \frac{12 \times 45 \times 110 \times 300}{8000 \times 315} = 7 \text{ inches approx.}$$

That is to say, the ship will rise $3\frac{1}{2}$ inches forward and sink $3\frac{1}{2}$ inches aft. Thus, final draft

	ft. ins.	ins.	ft. ins.
Forward	= 24 0	− $3\frac{1}{2}$	= 23 $8\frac{1}{2}$
Aft	= 26 0	+ $3\frac{1}{2}$	= 26 $3\frac{1}{2}$

Had the vessel in Fig. 19 been forcibly held at $W_1 L_1$ until the weight was placed at b, the moment of the mechanical couple then formed by the weight and buoyancy would have been $W \times GG_1$, or its equivalent $w \times h$; this is called the "moment to change trim," and its necessary value to produce a *given* change of trim can be estimated from equation (I). For, transposing that equation,

$$\text{the moment to change trim } (w \times h) \text{ is } = \frac{W \times GM_1}{12 \times L} \times x$$

and thus can be calculated when x is known.

To find the moment to change trim *one inch*, put $x = 1$. Then

$$\text{moment to change trim one inch is } = \frac{W \times GM_1}{12 \times L} \text{ foot tons.}$$

From this it is seen that ascertaining from equation (I) the change of trim in inches due to moving certain weights through known distances, consists in dividing the moment to change trim $(w \times h)$ by that necessary to change trim one inch.

Remembering the approach to equality of the length of war ships of usual proportions and their longitudinal metacentric height, the moment necessary to change trim one inch becomes approximately $= \frac{W}{12}$. This may be of use for rapidly determining whether any possible movement of weights on board a particular ship, of which only the displacement is known, will enable her to get over an otherwise impassable bar at the mouth of a river or harbour. For instance, suppose the ship mentioned in the preceding example can only pass over a particular bar when drawing not more than $25\frac{1}{2}$ feet of water aft, and that it is possible to move the following weights the given distances forward :—20 tons, 200 feet; 25 tons, 100 feet; 15 tons, 70 feet; and 30 tons through 50 feet. Will the movement of these weights be sufficient?

Since the ship has to rise 6 inches aft, there will be an equal alteration forward, so that the change of trim must be 12 inches. Now, the moment necessary to make this change is approximately $= \frac{W}{12} \times 12 = 8000$ foot tons.

And the moment to change trim provided by moving the weights is $20 \times 200 + 25 \times 100 + 15 \times 70 + 30 \times 50 = 9050$ foot tons.

Hence it may be concluded that by moving these weights the vessel may be taken over the bar. Of course, if the vessel is very long and of light draft, an allowance must be made for the difference between its length and the longitudinal metacentric height, as previously mentioned; this necessitates moving more weights on board than would be requisite under ordinary circumstances.

It may be remarked in passing, that the determination of the trim given to a ship by putting weights of moderate amount on board, may be reduced to the simpler case of moving weights already on board as follows :—The weights are supposed first placed in the same vertical line as the centre of gravity of the water plane area, in which position they will sink the ship deeper in the water *without change of trim.* This sinkage can be readily estimated when the tons per inch at the water line is known. Then the weights are supposed distributed to their proper places in the ship, and the consequent moment to change trim calculated in a manner similar to that adopted in the preceding example; observing that if some weights are moved forward and some aft, the *difference* of the moments forward and aft must be taken as the final moment to change trim. To take an example :—On a ship which has a draft of 25 feet forward and 27 feet aft, there are weights of 70, 90, and 20 tons to be placed. The first is to go 30 feet before the centre of gravity of the water plane area, and the second and third are to go 50 feet and 40 feet respectively abaft that centre of gravity. If the tons per inch at the load line is 40, and the moment to change trim one inch = 800, what will be the draft of water forward and aft, after the addition of the weights?

The total weight put on board is 180 tons, and therefore, supposing it so placed as to cause no change of trim,

$$\text{Sinkage} = \frac{180}{40} = 4\tfrac{1}{2} \text{ inches,}$$

and the draft would then be: Forward, 25 ft. $4\frac{1}{2}$ ins., and Aft, 27 ft. $4\frac{1}{2}$ ins.

If now the weights are supposed removed to their proper places,

$$\left.\begin{array}{l}\text{The moment tending to trim the ship}\\ \quad\text{by the stern is}\end{array}\right\} 90 \times 50 + 20 \times 40 = 5300$$

$$\left.\begin{array}{l}\text{The moment tending to trim the ship}\\ \quad\text{by the head is}\end{array}\right\} 70 \times 30 = 2100$$

$$\therefore \text{ Resultant moment trimming the ship by the stern} = 3200$$

$$\text{Whence change of trim} = \frac{3200}{800} = 4 \text{ inches.}$$

The final drafts required are, therefore,

	ft.	ins.		ins.		ft.	ins.
Forward -	25	$4\frac{1}{2}$	$-$	2	=	25	$2\frac{1}{2}$
Aft - -	27	$4\frac{1}{2}$	+	2	=	27	$6\frac{1}{2}$

Reverting now to Fig. 10, it will be remembered that when the ship is at any angle, the couple W × G Z is the *statical stability* at that angle. The weight of the ship being known, this stability can be calculated when the length of the arm G Z is known. For small angles, where the line of action of the buoyancy passes through the metacentre, G Z is equal to G M sin. θ, so that for angles to which this "metacentric method" applies, the statical stability is W $\times$ G M sin. θ foot tons. For larger angles than this, the lines of action of the buoyancy will not pass through the metacentre, and then to determine the values of G Z a different method must be pursued.

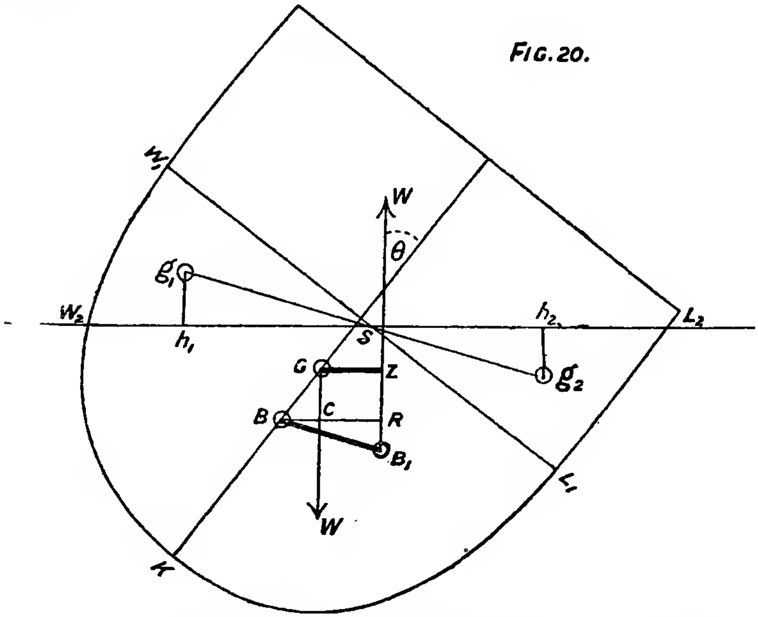


FIG. 20.

In Fig. 20, suppose W_1 L_1 to be the water line at which a ship floats when upright, B marking the position of the centre of buoyancy for that condition. Also suppose that after the vessel is inclined through a considerable angle, θ, the water line becomes W_2 L_2, the displacement remaining the same as before. Since these displacements are equal, the remainders after taking away the volume W_2 S L_1 K W_2, which is common to both, must be equal. That is, the wedge-shaped volume W_1 S W_2 (known as the wedge of *emersion*) must be equal in volume to the wedge L_1 S L_2 (the wedge of *immersion*), although of a different shape. Let g_1 be the centre of gravity of the wedge of emersion and g_2 that of the wedge of immersion, and join $g_1\,g_2$. It is seen that the displacement up to W_2 L_2 differs from that up to W_1 L_1 only in having the volume of the wedge W_1 SW_2 transferred from g_1 to g_2. Hence, if V is the volume of displacement, and v that of each wedge, the centre of buoyancy of the ship in the inclined position will be—by the principle already established—at B_1, where B B_1 is drawn parallel to $g_1\,g_2$, and is of a length given by

$$B\,B_1 = \frac{v \times g_1\,g_2}{V}$$

The line drawn through B_1 perpendicular to $W_2 L_2$ will be the new line of action of the buoyancy, and G Z, drawn at right angles to this from the centre of gravity, G, will be the required arm of the righting couple. To obtain the length of this arm, draw $g_1 h_1$ and $g_2 h_2$ perpendicular to $W_2 L_2$, and B R parallel to G Z. The distance $h_1 h_2$ is the *horizontal* distance through which the wedge is moved, and B R is the transfer of the centre of buoyancy in a horizontal direction. Hence, by an extension of the principle mentioned before, the length of B R is

$$\frac{v \times h_1 h_2}{V}$$

So that if C is the point of intersection of the line of action of the weight with B R,

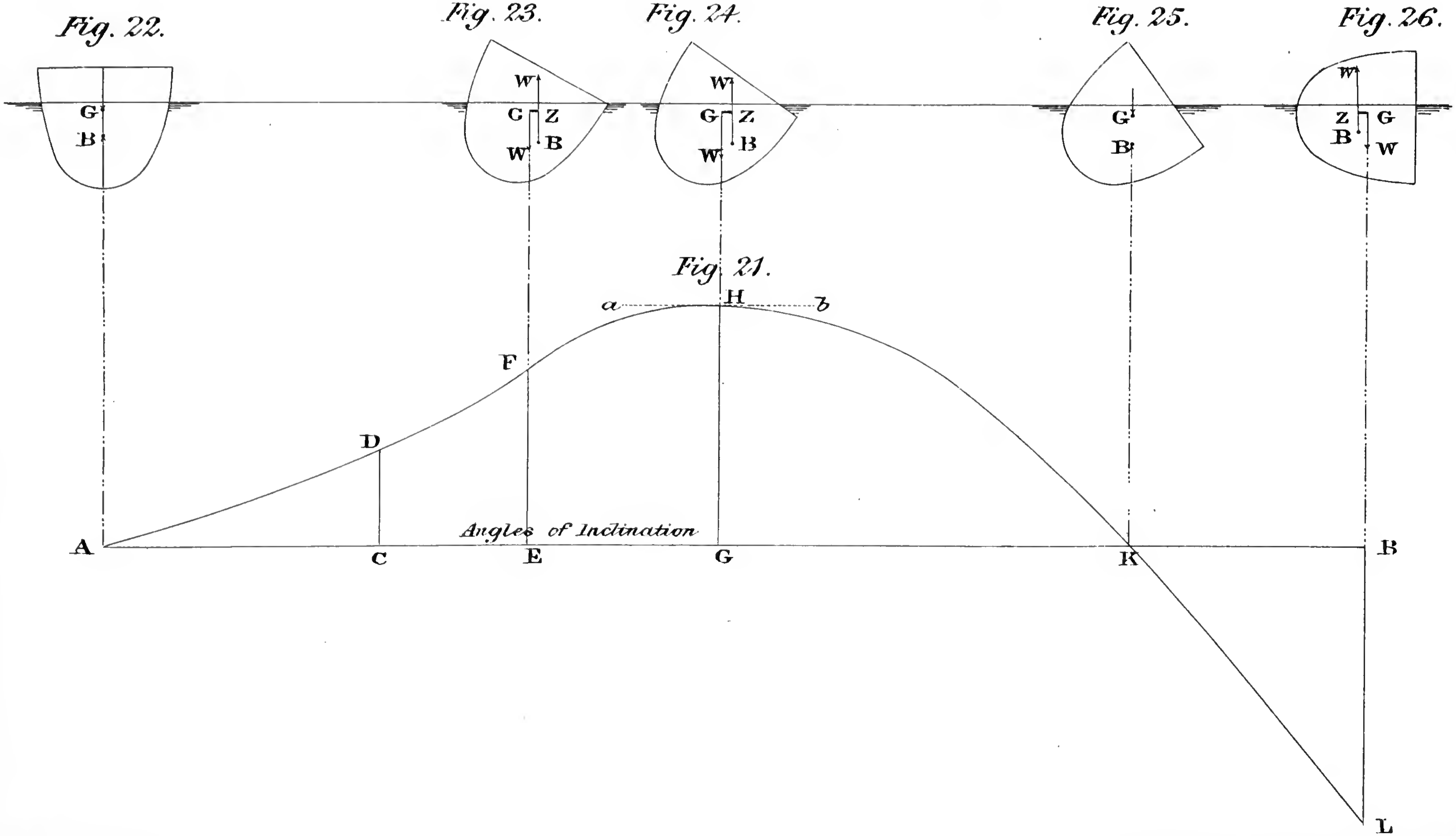
$$\begin{aligned} GZ &= CR = BR - BC \\ &= \frac{v \times h_1 h_2}{V} - BG \sin.\theta \end{aligned}$$

For any particular ship the positions of B and G are known, as well as the displacement, so that B G and V are known quantities in the above equation. Also, for any given angle of inclination *sin.* θ is known, and thus the determination of the value of G Z for any angle resolves itself into finding the value of the product $v \times h_1 h_2$ for that angle, a proceeding involving rather laborious calculation.

The length of G Z obtained as above varies, of course, according to the inclination; but as, for equal angles on opposite sides of the vertical, the values of G Z are obviously the same, those values need only be investigated for inclinations on one side. It should also be stated that when the angle through which a ship is inclined is considerable, the line of action of the buoyancy may fall to the left of G, Fig. 20, in which case the stability is said to be *negative*, since the weight and buoyancy form an *upsetting* instead of a righting couple.

The lengths of G Z are calculated for every 8 or 10 degrees of heel, until the stability becomes negative, and the results are shown on a *curve of statical stability* as shown on Plate I., Fig. 21. This is constructed in the following manner. A horizontal base line A B is taken, on which to set off angles of inclination, measuring from the point A, the scale employed being usually such that a quarter of an inch represents one degree. Let A C denote, on this scale, one of the angles for which the length of arm of righting lever has been found. From C draw C D at right angles to A B, and of such a length as to indicate to scale—a quarter of an inch usually representing one-tenth of a foot—the length of

Plate I.



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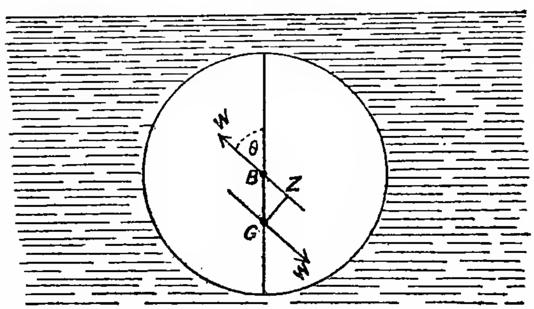
Judd & Co. Lith. 73 & 75, Farringdon Rd & Doctors' Commons 2134. 3.91.

G Z at the given angle. If, as is assumed, the couple formed by the weight and buoyancy at the angle denoted by A C tends to turn the ship towards the upright position, C D is set *above* the base A B as in Fig. 21; had the stability been *negative* at that angle, C D would have been set *below* A B. A similar construction being made for each of the other calculated values of G Z, a curve A D F H K L drawn through all the points such as D, is the required curve of statical stability, or, shortly, the *curve of stability*.

Fig. 21 shows the curve for the vessel indicated in Fig. 22, the upright position being supposed one of stable equilibrium. This exhibits the usual characteristics of such curves, and the varying position of the line of action of the buoyancy with regard to the centre of gravity can be traced by its means. In the upright position there is no arm of righting couple, the curve of stability meeting the base line at A. As the vessel is inclined, the line of action of the buoyancy falls to the right of G, so that the weight and buoyancy form a *righting* couple, indicated by the curve rising at first above A B. The value of G Z rapidly increases up to the angle denoted by A E, this being the angle at which the deck edge becomes immersed as depicted in Fig. 23. Beyond this, the increase of G Z is less rapid, and finally ceases, G Z attaining a maximum value at the angle represented by A G, where the tangent (*a b*) to the curve is horizontal. At this angle, therefore, the effort of the ship to right herself is greatest. As the inclination is further continued, the value of G Z diminishes, and the line of action of the buoyancy will eventually pass through G again as in Fig 25, in which case the vessel will be in equilibrium once more, the value of G Z being zero. Finally, beyond this second position of rest, the buoyancy will fall to the *left* of G, and will operate, together with the weight, to heel the ship farther from the upright position. This fact is indicated on the curve by its being below A B. Hence, if the ship is inclined by external forces to any angle less than that represented by A K, she will return to the upright when the external forces cease to operate. If, however, those forces incline the ship beyond that angle, she will overturn. The angle A K is known as the *angle of vanishing stability*, and its amount is the *range of stability*. This is one important criterion as to the probable safety of the vessel at sea. Other elements of the curve influencing the ship's safety are the maximum length of the arm of righting couple and the angle at which this occurs. Other things being equal, the greater the range of stability and maximum value of G Z—in other words, the greater the *area* of the curve of stability—the greater is the probability of the vessel's safety at sea. This area is therefore made as large as possible consistently with other conflicting conditions. For example, since for the first few degrees from the upright, the value of G Z is very approximately equal to G M *sin.* θ, it is seen that the greater the metacentric height is made in a ship, the steeper will the front part of the curve be, and the greater its area. But the adoption of a great metacentric height leads to heavy rolling at sea. So, in adjusting these clashing interests, the experience gained by successful ships is mainly relied upon; remembering the general principles that a mastless ship requires less stability than one carrying a large spread of canvas, and one with large resistance to rolling—due to bilge keels, &c.—less than one in which this resistance is small (*see* Chap. III).

It was remarked that a ship is in equilibrium when inclined at her angle of vanishing stability; that equilibrium is, however, evidently unstable, since, if the vessel is slightly moved in either direction from that position she will return to the upright, or overturn, according to the direction of inclination. It thus appears that whilst the points in which the curve of stability cuts the base line denote angles at which the ship will be in equilibrium, the *character* of that equilibrium is indicated by the relative positions of the curve and base line. If in passing towards the right from a position of rest the curve lies *above* the base, the equilibrium is *stable*; if *below* it, *unstable*.

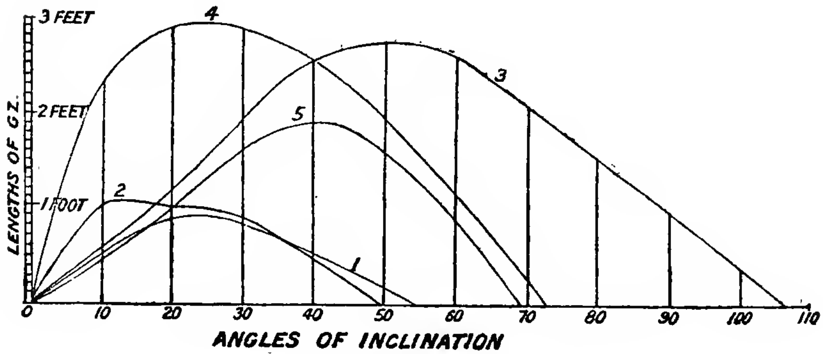
FIG. 27.



A class of vessel for which a curve of stability can be easily constructed is that designed for submarine service. Let Fig. 27 show a transverse section of such a one, having its centre of buoyancy for the upright position at B, and its centre of gravity at G. If the vessel is inclined, the shape of the displacement is not altered in the least, and therefore the centre of buoyancy for the displaced position is also at B, through which the new direction of the buoyancy acts. It is thus seen that, in a submarine vessel, the metacentre coincides with the centre of buoyancy, so that for stable equilibrium in the upright, *the centre of gravity must fall below* B. Again, since the buoyancy acts through B whatever the inclination, the arm of righting lever for any angle θ is equal to B G *sin.* θ; and as B G is known for any particular ship, and the value of *sin.* θ can be obtained from mathematical tables, the lengths of G Z can be readily calculated. The stability will be a maximum when sin. θ is greatest, and will vanish when *sin.* θ is zero. The former is the case when $\theta = 90$ degrees, *sin.* θ being then equal to 1; and the latter when $\theta = 180$ degrees (G being then vertically above B) as well as when the ship is upright. The equilibrium when $\theta =$ 180 degrees is evidently unstable, and on the slightest disturbance from that position the vessel will return to the upright. Such a vessel, then, has its maximum stability at 90 degrees, a range of stability of 180 degrees, and possesses the property of being self-righting, that is, of returning to the upright from any other position in which she happens to be. In passing, it may be mentioned that lifeboats have this quality of self-righting, the air spaces fitted along their sides and the considerable sheer (or curve upwards) given to their ends bringing the metacentre so low down, when the boat is keel upwards, that it falls below the centre of gravity, making that position one of unstable equilibrium.

In obtaining the curve of stability of an ordinary vessel, of the parts lying above the water only those which can be made watertight—that is, the parts contributing the reserve of buoyancy—are reckoned as assisting to give stability to the ship; and the following further assumptions are made. The centre of gravity of the ship is supposed to remain in the same position throughout, which assumes that no weights fetch away as the ship is inclined; and it is also taken for granted that no water enters into any watertight space either above or below the water line, through ports, scuttles, hatchways, &c., as the vessel is heeled over. On the first of these two latter assumptions Mr. White (*Manual of Naval Architecture*, page 123) remarks:—"This may be considered an improper supposition, especially in cases where stability is maintained beyond the inclination of 90 degrees from the upright; but it is to be observed that such extreme inclinations are not likely to be reached, whereas for less inclinations the supposition affects all classes similarly." And as to the second, "This assumption is fair enough as regards most of the openings, which are furnished with watertight covers, plugs, &c.; and as regards some of the hatchways which are usually kept open even in a seaway, it is only necessary to remark that they might be battened down on an emergency, while their situation near the middle line of the deck prevents the water from reaching them except at very large angles of inclination." (pp. 123–4). On these assumptions the curves shown in Fig. 28 for four of H.M. ships and the American ironclad monitor *Miantonomoh* have been obtained. A considera-

FIG. 28.



References.

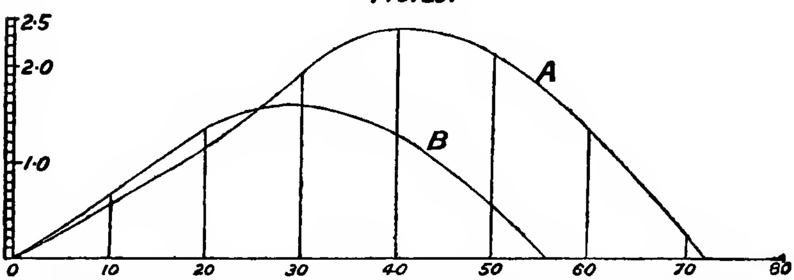
1. *Captain.* 2. *Glatton.* 3. *Inconstant.* 4. *Miantonomoh.* 5. *Monarch.*

tion of these curves will bear out the statement previously made that the steepness of the front part is dependent upon the metacentric height. That for the *Miantonomoh* is steepest, this vessel having a metacentric height of about 14 feet. Of the others the *Captain* had a height of 2·6 feet, and the corresponding information for the remainder has been previously given (page 23).

A comparison of the curves for the *Captain* and *Monarch* exhibits the great influence of *freeboard* or height from the water line of the upper deck amidships at the side, on the area and range of curves of stability. The displacements of these two ships only differ by 4 or 5 per cent., and their proportions are not greatly different, the great variation in the two designs being in the matter of freeboard. The *Monarch* has 14 feet, whereas the *Captain* possessed 6½ feet only. In consequence of the slightly smaller metacentric height of the former, her curve of stability falls at first below that for the *Captain*; but she has a total range of stability of 69½ degrees, reaching her position of maximum stability at 40 degrees—where the G Z is 21¾ inches—whereas the *Captain* has a total range of 54½ degrees only, and reaches her position of maximum stability at 21 degrees, at which angle she has a G Z of only 10¾ inches. The reason for this influence of freeboard is not difficult to understand if reference is made to Fig. 21. For the greater the freeboard, the greater will be the angle at which the deck edge becomes immersed and the farther will the portion A D F of the curve be prolonged before that curve begins to turn round towards the base as at F. Of the other vessels, the curves of which are given above, the *Inconstant* has a freeboard of 15¼ feet, the *Miantonomoh* 3 feet, and the *Glatton* also 3 feet, although this last ship has in addition a superstructure on the upper deck amidships not extending to the side, a fact which accounts for the irregularity in the curve of that vessel. It must also be remarked that in comparing ships of greatly different proportions, the freeboard of each ship must be viewed with reference to its breadth; it being evident that 10 feet of freeboard on a ship 50 feet broad will be more influential than the same amount in one 70 feet in breadth, on account of the larger angle to which the former must be inclined before the deck edge becomes immersed. The freeboard must also be compared with the *total depth*, a given amount having a greater relative effect in a shallow than in a deep ship.

The comparison instituted between the *Monarch* and *Captain* serves to indicate the great influence which reserve of buoyancy—of which freeboard is to some extent a measure—has upon the curve of stability, but one or two further illustrations may not be without value. For example, in vessels of the *Canada* class, if the reserve of buoyancy is increased by making the poop and forecastle watertight, the range of stability is increased by about 16 degrees from its value when those erections are open; and the area of the curve is also enlarged. Again, the curve marked A in Fig. 29 is the curve of stability for a vessel having a superstructure extending from side to side above the upper deck for rather more than one-third its length—a vessel somewhat similar, in fact, to the one shown in Fig. 51. If the superstructure is thoroughly destroyed by shell fire, the material falling upon the upper deck

FIG. 29.



will cause the centre of gravity of the ship to move downwards, and will thus increase the metacentric height. At first, therefore, the stability of the damaged ship compares favourably with that for the intact condition, but in consequence of the decrease in the reserve of buoyancy, the area and range of the curve of stability is lessened to the extent shown by the curve B. Further, if the reserve of buoyancy could be *totally* destroyed in a vessel, a simple consideration will show that she must inevitably capsize. For, imagine a complete watertight deck to be worked at the height of the water line, and the part above this demolished. The vessel would still continue to float, since the buoyancy remains uninjured, but, as she is now in the condition of a submarine vessel with the watertight deck awash, her metacentre will be at the centre of buoyancy, the position of which latter point has, of course, not been altered by the damage above water. Now in ordinary vessels the centre of buoyancy is several feet *below* the centre of gravity, and therefore the ship under consideration is in unstable equilibrium when upright, so that, like other submarine vessels in similar circumstances, she must capsize.

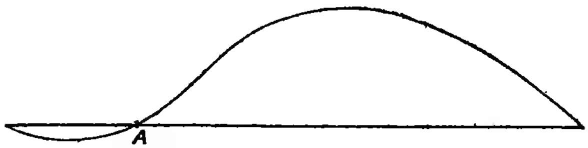
The hypothetical deck mentioned above very closely resembles the protective deck fitted in many ships; and although of course it would be practically impossible for a vessel to become reduced to the above condition, looking at the matter in this light enables the importance of the reserve of buoyancy to be better recognized, and shows the great necessity for protecting against too extensive damage that part of the ship lying above water. This protection may be given by armouring the sides; by minutely subdividing the space above water so that any damage may be localised; by stowing stores in that space, in order that, even after it may be laid open to the sea, a reserve of buoyancy equal to the volume of the stores may still be retained; or, finally, by any combination of the others. These arrangements will be noticed in Chapter XIV.

Curves of stability are not only constructed for the fully laden condition, but also for the extreme light condition of the ship. When the stores are consumed the vessel has additional freeboard due to her rising in the water; this fact tends to give to the curve for the light condition greater area and range than that for the load condition. On the other hand, the metacentric height is usually less for the light condition, leading to a reduction in the size of the curve for that condition. Of these two opposing tendencies, the latter is the more potent, generally speaking, the curve for the light condition having less area and range than that for the load condition, although the reduction is not great.

It has before been remarked that some merchant ships in the light condition are unstable when upright. Such vessels will have a curve of stability for that condition similar to the one shown in

Fig. 30, the point A, where the curve begins to rise above the base line, denoting the angle to which the ship will loll to one side or

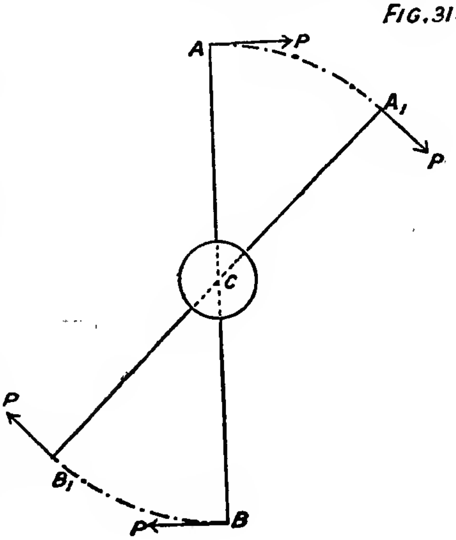
FIG. 30.



other of the vertical before attaining a position of stable equilibrium. After passing this angle, the vessel will usually have a good range of stability, so that it can by no means be inferred that an ordinary ship can be easily capsized if her upright position happens to be one of unstable equilibrium.

Attention will next be directed to what is known as *dynamical* stability. The amount of mechanical *work* necessary to heel a ship over to any angle from the upright position, is called the *dynamical stability* at that angle. Work is said to be done when a resistance is overcome through space, and is usually expressed in foot-tons when dealing with large amounts. For instance, in raising a weight of one ton through a vertical distance of one foot, one foot-ton of work is performed ; if through two feet, two foot-tons, and so on, the amount of work always being estimated by multiplying the resistance by the distance through which it has been overcome. The same term "foot-ton" has been used with a different meaning when speaking of *statical* stability, and the nature of the difference may be illustrated by a simple example. In Fig. 31 let A and B be the ends of two capstan bars, at each of which a man exerts a pressure P in the directions shown, when just balancing the force acting upon the body of the capstan. The product P × AB measures the effect of the *couple* exerted by the men in foot-tons, if P is expressed in tons and A B in feet, and corresponds to the expression (W × G Z) for the *statical* stability of a ship. So long, however, as the men stand still at A B no resistance is overcome

FIG. 31.



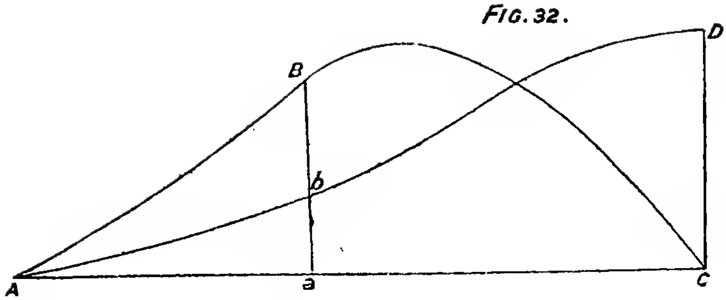
through space, and hence no work is performed. Suppose now the men, each steadily exerting the force P, move the bars from A B to A_1 B_1. In the new position, the magnitude of the couple exerted by the men is still P×A B foot-tons, but the *work* done in foot-tons in going to that position differs altogether from this. The amount accomplished by the man at A is P multiplied by the length of the circular arc A A_1, and by the one at B is P multiplied by the length of the arc B B_1. The length of the arc A A_1 is equal to A C multiplied by the circular measure of the angle A C A_1, and B B_1 is obtained by multiplying B C by the circular measure of the same angle. Hence

$$\begin{aligned}\text{Total work done} &= P \times (A\,A_1 + B\,B_1) \text{ foot-tons} \\ &= P \times (A\,C + C\,B) \times \text{circular measure of the angle } A\,C\,A_1 \\ &= (P \times A\,B) \times \text{circular measure of the angle } A\,C\,A_1.\end{aligned}$$

It is thus seen that the work done in foot-tons by a couple turning through a certain angle is equal to the *moment* of the couple in foot-tons multiplied by the circular measure of that angle.

In the case of a ship, if fluid resistance be neglected, the work done in inclining her to any angle is expended in overcoming the resistance offered at each instant by the statical stability, and differs from the preceding example only in the fact that the moment to be overcome varies from instant to instant in this case, owing to the variation in the length of the arm of righting couple. To investigate the amount of work done during the inclination of a ship, a modification of the method adopted in the above example is resorted to, by which means it may be shown that *the area of the curve of statical stability up to any angle, measures the dynamical stability at that angle.* For example, if A B C, Fig. 32, is the curve of statical stability for a ship, the area of the part A a B, when properly interpreted, will give the dynamical stability for the angle A a. If the distance $a\,b$ is set up to represent on a certain scale the area A a B, and a similar construction followed out for several other angles, the curve A b D drawn through all such points as b, is the *curve of dynamical stability*, which gives for any angle the dynamical stability at that angle by simple measurement. From the foregoing it is seen that the greater the area of the curve of statical stability, the greater will be the amount of work necessary to be done on the ship to incline her to the angle of vanishing

FIG. 32.



stability, and the less likely therefore is she to be inclined to that critical angle by the action of external forces. Thus is demonstrated the importance of the *area* of the statical stability curve in relation to the safety of the vessel.

Brief reference will now be made to the effect on the stability of damage to a ship at or below the water line. If the injury occurs above a watertight flat, so that the surface of the water is at the same height inside as outside the ship, the area of the intact water plane is thereby decreased, and the distance between the centre of buoyancy and metacentre thus shortened (*see* page 21); on the other hand, as the ship sinks in the water, consequent upon the damage, the centre of buoyancy rises, and thus the final position of the metacentre and value of the metacentric height (which determines the stability at small angles) is dependent upon the relative effect of loss of water plane area and rise of centre of buoyancy. As an extreme example, reference may be made to the *Inflexible.* If her unarmoured ends are completely riddled at the water line, her metacentric height falls from $8\frac{1}{4}$ to 2 feet, indicating the importance of protecting the water plane area from excessive damage, and of providing vessels of the same type as *Inflexible* with greater metacentric heights than would otherwise be necessary.

If, however, the damage is low down, below a watertight flat, so that the compartment is completely filled, it will make no difference to the stability if the damaged part is afterwards made good. Conceive this to be the case, the admitted water being virtually so much ballast low down in the ship. This will therefore considerably lower her centre of gravity, whilst the sinkage due to the admission of the water will but slightly affect the position of the metacentre. Such injury will thus increase the metacentric height, and, although the freeboard has been somewhat diminished, the area and range of the curve of stability will be little if at all diminished from their values when the vessel is intact.

If water in the hold of a vessel is free to shift, the stability at any angle is less than it would have been had the water been fixed in position, because, in travelling towards the side to which the vessel is inclined, the water carries the centre of gravity of the whole ship nearer the line of action of the buoyancy.

Examples.

I. The water lines of a vessel are 3 feet apart, and the depths below the respective lines of the corresponding centres of buoyancy are 10·5, 9·0, 7·5 and 6·1 feet, beginning with the load water line. The distances between these centres of buoyancy and their corresponding metacentres are, respectively, 12·3, 13·9, 16·0, and 19 feet. Construct the metacentric diagram to scale.

II. An inclining experiment, performed on the ship of the preceding question when floating at a line 4 feet below her load water line, showed that in this light condition, a weight of 31 tons moved 40 feet across the vessel from side to side inclined her to an angle the tangent of which was $\frac{1}{12}$. The weight of the ship at the time of the experiment (including inclining ballast) was 5,952 tons; what was her metacentric height at that time?

Again, if in putting the necessary weights on board to sink the vessel to her load water line from the light condition, her centre of gravity is lowered 6 inches from its position in that condition, what is the metacentric height of the vessel when fully laden?

Answers—2·5 feet; 2·8 feet.

III. A weight of 68 tons, already on board a vessel 315 feet long and of 7,497 tons displacement, is moved forward through a distance of 105 feet. What change of trim is thereby produced, the longitudinal metacentric height being 300 feet?

Answer—12 inches.

IV. If before moving the weight in question III. the draft of water had been $23\frac{1}{2}$ feet forward and $25\frac{1}{2}$ feet aft, what would be the new drafts after that weight was moved?

Answer—24 feet forward; 25 feet aft.

V. The weights enumerated are to be placed on board a ship drawing 25 feet of water forward and 26 feet aft, in the positions specified relatively to the centre of gravity of the load water plane ; viz.,· 30 tons 20 feet before and 40 tons 30 feet before ; also 50 tons 70 feet abaft and 20 tons 90 feet abaft. Assuming the tons per inch to be 35, and the moment to change trim one inch 700, what will be the new draft of water forward and aft ?

Answer—25 feet 1½ inches forward ; 26 feet 6½ inches aft.

VI. Construct to scale the curve of stability of which the ordinates at intervals of 10 degrees are, starting from the upright, 0, 1·09, 2·3, 2·98, 2·72, 1·84, ·57 and — ·89 feet respectively. Also, find (1) the range of stability and (2) the maximum length of arm of righting couple and the angle at which it occurs.

Answers—64 degrees ; 3 feet ; 32 degrees nearly.

CHAPTER III.

OSCILLATIONS OF SHIPS.

The principal oscillations of ships take place in the transverse and fore-and-aft directions ; in the former case the motion is termed *rolling*, and in the latter *pitching* or *'scending* according as the bow of the ship moves downward and the stern upward, or *vice versâ*. Of these oscillations rolling is much the more important, and will alone be dealt with here.

The extent of the rolling motion is measured by the inclinations which the ship reaches from the upright, and in going from her extreme position on one side of the vertical to the next extreme position on the opposite side, she is said to perform a single oscillation. The *arc of oscillation* is the total angle swept through by the ship in an oscillation, and her *period* is the time occupied in performing one oscillation. For example, the *Devastation* was found to move from an angle of 10 degrees on the starboard side to 8 degrees on the port side in 6·76 seconds ; her arc of oscillation at that time was therefore 18 degrees and her period 6·76 seconds.

The main object in studying this branch of Naval Architecture is to ascertain what elements govern or affect the rolling motions of a vessel in a seaway, so that her probable safety at sea may be secured. But in approaching this question, although it is evident at once that a ship will experience resistance when rolling through the water, it is convenient to consider *first* the purely hypothetical case of a vessel rolling *unresistedly* in still water, where the only two forces acting upon her are the equal ones due to her weight and buoyancy ; *secondly* to enquire, by means of experiments on actual ships or models in still water, in what respects the conclusions arrived at in the above imaginary case are modified by the operation of resistance ; and *lastly* to see in what manner the oscillations are affected by the presence of waves. This order will therefore be preserved in the following remarks.

First then, consider a ship rolling in still water subject only to the two forces of the weight and buoyancy. Suppose her forcibly inclined to the position shown in Fig. 34. To do so requires the

FIG. 34.

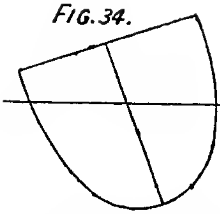


FIG. 35.

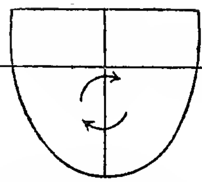
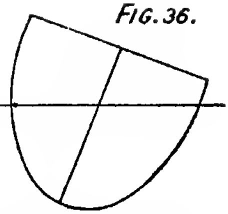


FIG. 36.



expenditure of *work* or *energy*, the amount of which can be ascertained from the curve of dynamical stability, and which could be again recovered from the vessel if she were attached to, say, suitable pulleys and thus forced to raise a weight when released from her inclined position. She has, in fact, acquired energy *by virtue of her position* from the vertical. Now by a well known principle —that of conservation of energy—if no external resistances act upon a body, the energy due to position added to that due to motion remains a constant quantity. Applying this to the case in question, suppose the unresisted vessel suddenly released from the position Fig. 34. At that instant there is no motion, and consequently her total energy is the dynamical stability in that position. Reaching the upright, Fig. 35, she has no energy by virtue of position, all the dynamical stability having been converted into energy of motion. Finally, on coming to rest on the opposite side of the vertical, Fig. 36, all the energy she possesses is that due to position, so that the dynamical stability there must be equal to that she had when first released. That is to say, the ship must reach, on the opposite side of the vertical, an angle equal to that from which she first started. If, therefore, a vessel rolled unresistedly in still water, she would continue to oscillate from side to side without any diminution in the angle of roll from the perpendicular.

Again, assuming that the front part of the ship's curve of stability is a straight line (an assumption which is approximately true in most ships up to 10 or 15 degrees), a mathematical investigation shows that the *period* is the same for large as for small arcs of oscillation ; that is, she will swing through a total arc of, say 20°, in the same time as through one of, say 4°. This fact is usually expressed by saying that the ship is *isochronous* within the limits of roll mentioned above. If the period be denoted by T, its value in seconds is given by the formula

$$T = \cdot 554\sqrt{\frac{k^2}{m}} \text{ or } T = \cdot 554\frac{k}{\sqrt{m}}$$

Where m is the metacentric height, and k the *radius of gyration* when the ship is oscillating about a longitudinal axis through her centre of gravity. To explain the meaning of the term *radius of gyration*, a definition of *moment of inertia* will be necessary. Conceive the ship to be made up of a very large number of small parts or elements ; then the moment of inertia, about a longitudinal axis through her centre of gravity, is the sum of the products obtained by multiplying the weight of each element by the square of its perpendicular distance from the axis. For example, if w_1, w_2, w_3, &c., denote the weights of elements situated at distances of x_1, x_2, x_3, &c., respectively from the axis, the moment of inertia is

$$w_1 \times x_1^2 + w_2 \times x_2^2 + w_3 \times x_3^2 + \&c.$$

The total weight of the ship is, of course, the sum of the weights w_1, w_2, w_3, &c. ; let this be represented by W, and the moment of inertia by I. Then the *radius of gyration* (k) is such a length that

$$Wk^2 = I.$$

In other words, if the total weight of the ship could be concentrated at a point distant from the axis a length equal to the radius of gyration, it would have a moment of inertia equal to that which the ship actually possesses about the same axis.

To take an example of the above formula, the *Cyclops* class have a radius of gyration of about 18 feet and a metacentric height of 3·45 feet. Hence their time of performing a single oscillation, if rolling unresistedly, would be given by

$$T = \cdot 554 \sqrt{\frac{324}{3 \cdot 45}} = \cdot 554 \times \frac{18}{1 \cdot 86} = 5 \cdot 36 \text{ seconds.}$$

The usefulness of this formula will be noted farther on.

For very large angles of roll, the period will be increased from that given by the formula for isochronous rolling, this being analogous to the behaviour of a simple pendulum. If a pendulum is of such a length as to have a period of one second when oscillating through small arcs, the period becomes increased to 1·073 seconds when the pendulum oscillates 60° on each side of the vertical, and to 1·762 seconds when swinging through 150° on each side.

Secondly: An actual ship set rolling in still water differs from the preceding case in that *fluid resistance* is in operation, in addition to the moment of stability. This resistance is due :—

(1) To the rubbing against the water of the surface of the ship when rolling, producing *frictional* resistance.

(2) To the opposition offered to the passage through the water, in a more or less flat-wise direction, of projections such as the

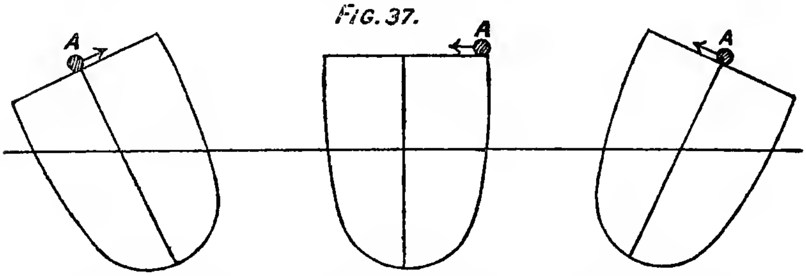
keel and bilge keels, and of the comparatively flat parts of the ship at the ends : this is known as *direct* or *head* resistance.

(3) To the creation of waves as the vessel rolls : these cannot be propagated without the expenditure of energy, which must be supplied to the water by the ship, thus affecting her motion.

The first is analogous to the resistance offered to the passage of a thin board edge-wise through water; and the second, to that offered by the water to the same board moved flat-wise through it.

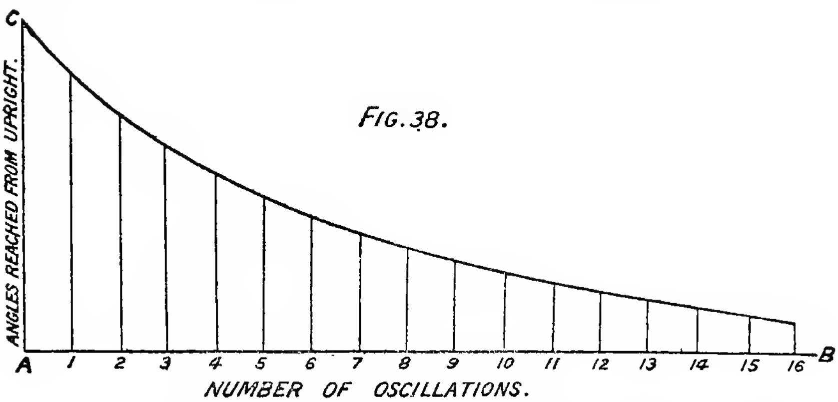
Before tracing the effects of fluid resistance, the methods employed to set ships rolling in still water will be noticed. The first, adopted for small ships, is similar to that employed at Brest in carrying out experiments on the *Elorn*, a vessel of about 100 tons displacement. The ship was placed parallel to the quay and at some little distance from it, and then a purchase was led in a horizontal direction from a pair of shear legs erected on board (for want of a mast) to a winch on shore, by which means the vessel was inclined to the required angle and then set free. Large vessels are set rolling through the action of a number of men on board, who run from one side of the deck to the other, their movements being carefully timed with reference to the motion of the ship. The men should always be on the descending side of the vessel, so that their weight may help to incline her further, and to accomplish this, their movements should be as follows. To start the inclination they must stand on one side of the deck and then run towards the middle line, reaching it at the time the maximum inclination of the ship occurs ; they should then run up the deck to the opposite side, stay there as long as possible and get back again to the middle line by the time the ship reaches her extreme angle on that side of the vertical. Thus the positions of the men and the directions in which they should run in relation to the positions of

FIG. 37.



the ship are as shown in Fig. 37, at A, from which it is seen that the men should, where possible, *always be running up an incline.* By this means large ships may be rolled several degrees from the vertical.

If, after a vessel has been set rolling by either the above methods, careful observations are made of the successive angles reached from the vertical, it will be seen that instead of these angles remaining always the same—as they would do in unresisted rolling—the effect of resistance is to continually diminish them, and finally to bring the ship to rest. The reason for this is apparent, for, when a vessel is held to a certain angle on one side of the vertical, it has at that angle an amount of dynamical stability or energy of position which can be obtained as before from the curve of dynamical stability ; but after she is released a portion of this energy is used up in overcoming the various opposing resistances, so that she will come to rest at such an angle on the opposite side of the vertical as will make the difference between the dynamical stabilities at the beginning and end of the oscillation equal to the energy absorbed by the resistances ; the greater the resistances, therefore, the greater will be the degradation of roll, and the sooner will the vessel be brought to rest. The successive angles reached on opposite sides of the vertical during a rolling experiment are carefully noted, until the inclination becomes reduced to two or three degrees, and the results plotted off on a *curve of declining angles*, or, as it has until recently been called, a *curve of extinction*. This



is constructed in the following manner. At one end, A, of a base line A B, a line A C is drawn at right angles to that base, to represent on a certain scale the extreme inclination attained by the ship at the instant of commencing observations. Equal distances are set off along A B to denote the number of oscillations ; and the observed inclination of the ship to the upright, at the end of each oscillation, is set up from the base line at the point representing that oscillation. Thus if the angle A C was reached on the starboard side, the angle set up at the position marked 1 is that next attained on the port side, and at position 2 the succeeding angle reached on the starboard side, and so on. A curve drawn through all points such as C is the curve required. The difference between any two successive ordinates gives the degradation of roll or *extinction value* in that particular oscillation, and the greater the resistance the greater will be this extinction value, and the steeper therefore the curve of declining angles.

If a model of a ship be made of reasonable size, and weighted so that it may have a time of oscillation proportioned to the period of the full sized ship, experiments on different sized models lead to the conclusion that they produce the same curve of declining angles as the ship, so that if a vessel and her properly constructed model be set oscillating from the same initial angle, they will have the same extinction value for corresponding oscillations and will thus be brought to rest after the same number of rolls. This is an important fact, for it shows that rolling experiments on a model may be substituted for those on the full sized ship, not only leading to the easier performance of the experiments and to the possibility of trying the behaviour of a model in conditions to which it would not be possible or convenient to reduce the ship (as for example, testing her resistance to rolling when damaged), but also enabling the resistance to rolling of a new design to be found and the results consequent upon any proposed modifications determined, before the building of that vessel is decided upon, or completed.

The difference between the dynamical stabilities at the beginning and end of an oscillation is, as has been stated, the energy absorbed by the action of the various resistances during that oscillation; and in first approaching the question as to the causes operating to produce the reduction of roll, the late Mr. W. Froude—to whom most of the recent advance in this subject is due—assumed them to be frictional and head resistance only. But by *calculating* the amounts of these two resistances, as he was able to do by means of *data* obtained from independent experiments, and comparing the loss of energy from these causes with the actual loss shown by the decreased amplitude of roll, a great discrepancy was apparent, which led Mr. Froude to seek for some further cause of loss and to find it in the wave making action of the ship. For example, H.M.S. *Greyhound* when released from an angle of 6° on one side of the vertical reached an angle of 5·65° on the opposite side. The difference between the dynamical stabilities at these two angles was found to be 4700 foot-pounds, which was therefore the energy abstracted by all the resistances. A very liberal estimate of the frictional and head resistances led to the conclusion that they certainly did not exceed 120 foot-pounds and 700 foot-pounds respectively, thus accounting for only 820 out of a total of 4700 foot-pounds. The remainder was absorbed in the creation of waves, Mr. Froude calculating that a wave 1¼ inches high would fully account for this outstanding difference. Results obtained in a similar manner for other vessels, also released at 6°, are given in the annexed table :—

Name of ship.	Energy absorbed by—		
	Friction.	Head Resistance.	Waves.
Sultan - - -	354	5,036	14,676
Inconstant - -	143	4,060	17,367
Volage - - -	96	2,944	11,047

From these figures it is seen that for small angles of roll, wave resistance is by far the most powerful agent in extinguishing the oscillations. As however the frictional and head resistances grow more rapidly with increase of angle than that due to waves, the former become more important at larger angles.

Observations of the *times* taken to perform the successive arcs of oscillation show that they are all equal, so that a vessel rolling *resistedly* is also isochronous for moderate angles of roll. Thus the period for a single oscillation can be obtained by observing the time taken to perform any number, and then dividing that time by the number. Again, rolling experiments also establish the fact that although adding resistances to a ship has a most marked effect in reducing the angle of roll, it has little effect upon the *period*. This is well illustrated by some experiments carried out by Mr. Froude on a model of the *Devastation*. This was released from 8½ degrees in each experiment, the only difference being that bilge keels of different sizes were fitted to represent, on the proper scale, the depths for the actual ship given in the accompanying table. The number of double rolls (starboard to port and back again to starboard) performed before the model was brought practically to rest was noted in each condition, as well as the period, with the following results :—

Conditions.	Number of Double Rolls before model was reduced to rest.	Period for Double Roll in seconds.
1. No bilge keels fitted ...	$31\frac{1}{2}$	1·77
2. Single 21-inch bilge keel each side.	$12\frac{1}{2}$	1·90
3. Single 36-inch bilge keel each side.	8	1·90
4. Two 36-inch bilge keels each side.	$5\frac{3}{4}$	1·92
5. Single 72-inch bilge keel each side.	4	1·99

Thus, taking the extreme case, whilst the increased resistance due to fitting bilge keels was such as to reduce the number of rolls from $31\frac{1}{2}$ to 4, the period was only altered ·22 seconds in the direction of lengthening it.

It may thus be inferred that as resistance has so little influence on the time of oscillation, it will be nearly the same as for *unresisted* rolling ; and this inference was shown to be correct by the French experiments on the *Elorn.* To get her period when rolling unresistedly, she was placed in dry dock and oscillated on suitably curved supports under her keel, the result thus obtained being 2·03 seconds. When rolled in water without bilge keels her period became 2·18 seconds, and this was increased to 2·33 seconds when relatively large bilge keels were added.

Hence the time of oscillation for a vessel rolling *resistedly* in still water is given, very nearly, by

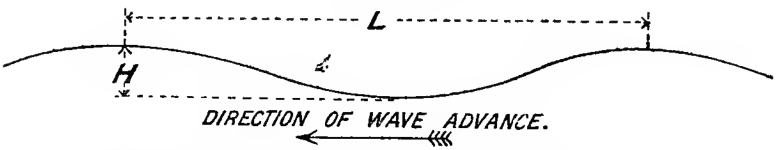
$$T = \cdot 554 \sqrt{\frac{k^2}{m}} \text{ or } = \cdot 554 \times \frac{k}{\sqrt{m}}$$

This formula is thus of use in showing that the period may be *increased*—and the ship therefore made to move more slowly—by increasing the radius of gyration or diminishing the metacentric height ; and may be *decreased* by decreasing the radius of gyration or increasing the metacentric height. The value of k depends upon the distribution of the weights relatively to the axis through the centre of gravity ; the greater their distance from that axis, the greater will be the value of the radius of gyration. This distribution, however, is not much under the control of the designer, so that the period for a ship of new design must be principally affected by the Naval Architect through the metacentric height. Vessels having small metacentric heights will roll slowly and *vice versâ.* For example, the *Inconstant* with a metacentric height of 2·8 feet has a period of 8 seconds, whilst that of the *Inflexible,* with a large metacentric height ($8\frac{1}{4}$ feet), is only 5·35 seconds, notwithstanding the fact that she has a larger radius of gyration.

Thirdly: The motion of a ship in a seaway must next be considered, but before doing so a few remarks will be made and definitions given respecting deep sea waves. In the first place, it is most important to remember, when speaking of the *speed* of a wave that it is only the *wave form* which advances, and not the water composing that wave. This is seen to be true by watching the behaviour of, say, a piece of wood amongst waves; it is seen to simply move backward and forward about a fixed mean position instead of travelling continuously in the direction of wave advance, as it would do if the mass of the water composing the wave moved in that direction. This motion of wave form as distinguished from that of the materials composing the wave may be illustrated by securing one end of a rope and giving a rapid up-and-down motion to the other end by means of the hand; a wave *form* will travel from one end to the other, but it is evident that the particles composing the wave have not so travelled. Another illustration is supplied by the wave form which traverses a field of corn from end to end when a wind is blowing.

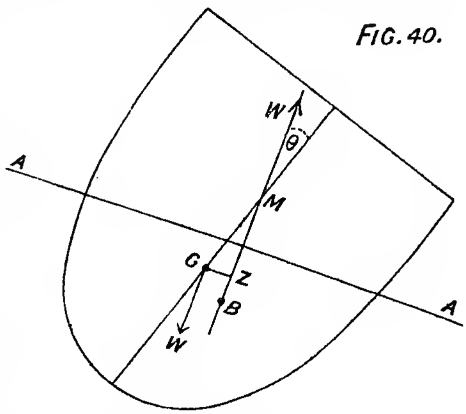
The *length* of a wave is the perpendicular distance between successive crests or hollows, and its *height* is the vertical distance from hollow to crest. Thus in Fig. 39, which shows a section of the wave at right angles to the crests, L is the length and H the height of wave.

FIG. 39.



If an observer, when stationary, notes the time which elapses between the passage past him of successive wave crests, he thereby ascertains the *period* of the wave. If he proceeds in the direction in which the waves are advancing, a longer time will elapse than before between the transit of successive wave crests; that is, the period of the waves has been apparently lengthened by this proceeding. On the other hand, travelling in a direction opposite to that of wave advance leads to an *apparent* period shorter than the real period. Similarly, proceeding in any oblique direction affects the apparent period; retreating obliquely from the waves *increases* and advancing obliquely towards them *diminishes* the apparent, as compared with the real, period.

The fundamental difference between still- and wave-water consists in the fact that, whilst the resultant fluid pressure upon a vessel immersed in the former always acts vertically upwards, that consequent upon immersion in the latter *acts at right angles to the surface of that part of the wave in which the vessel is floating.* This is only strictly true when the ship is extremely small relatively to the wave, but in assuming its truth for ordinary vessels the error made is on the side of safety. As a wave passes beneath a ship, therefore, the direction of action of the buoyancy is continually changing; when the crest reaches her it acts vertically upwards; about midway between crest and trough it acts at its maximum deviation from the vertical, and at the trough its direction is again vertical. The *amount* of the buoyancy in wave-water is also constantly varying according to the position of the ship on the wave, being sometimes greater and at other times less than her buoyancy in still water. The causes producing this variation also operate to make the *virtual weight* of the *ship* change in the same manner, as well as to make it also act at right angles to the surface of the wave. These differences in amount of the virtual weight and buoyancy are not, however, relatively large and are usually neglected, the virtual weight being taken to be her real weight throughout, and the buoyancy also assumed to be constantly equal to this. It is thus seen that the ship at each instant is acted upon by two equal and opposite forces in a direction perpendicular to the wave surface, and hence, if her masts happen to stand at right angles to the wave, she will at that instant be subject to no forces displacing her from that position. She is then, indeed, in the same condition as she would be if upright in still water, and therefore, in wave water, the perpendicular to the wave surface is the *virtual upright*; and if the vessel were displaced through an angle θ from that virtual upright, the moment of stability acting upon her would be equal to that acting upon the same vessel *in still water* when at the angle θ from the upright. In Fig. 40, let A A be the wave surface where a ship is floating at any particular instant; then if the vessel had its mast standing in the direction B M, perpendicular to A A, there would be no moment of stability; but in



the position shown, its mast making an angle θ with B M, the righting couple is W $\times$ G Z or W $\times$ G M sin. θ. It is thus clear that the stability, instant by instant, depends upon the inclination of the ship to the virtual upright, and if this becomes equal to or greater than the angle of vanishing stability, as given by the curve of stability, the vessel will capsize.

Bearing the above principles in mind, a mathematical investigation can be made of the behaviour of a ship rolling *unresistedly* amongst waves, and a comparison afterwards instituted between the results so obtained and the performance of actual ships, to ascertain the effect of resistance in modifying those results. Assuming the vessel to be placed broadside to, and acted upon by, a single series of deep-sea waves, all having the same period and dimensions, and neglecting resistance, investigation points to the following conclusions :—

(1) If a ship happens to fall in with waves having a period twice that of her own *natural* or still-water period, she will infallibly capsize after the passage of a few waves, no matter how great her angle of vanishing stability may be.

(2) If the still-water period is small compared with the wave period, the vessel tends to keep her deck parallel to the surface of the wave.

(3) If the period is long in comparison with that of the wave, the vessel will tend to keep upright, her deck not departing far from a horizontal position : and the longer the period of the ship the more nearly vertical does the vessel tend to remain. The reason for this is not far to seek, for if the vessel is at one instant so placed on the slope of a wave that she is moved away from the upright, she will not have had time to travel far before the opposite slope of the wave will be underneath her, urging her back again towards the vertical position. This quality of keeping approximately upright amongst waves is known as *steadiness*.

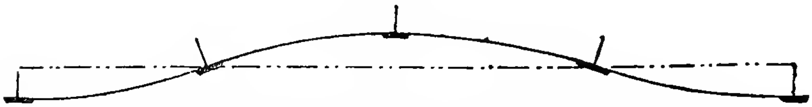
As before noticed, the perpendicular (or *normal*) to the wave surface is upright at a crest, passes through its extreme deviation from the vertical at about the mid-height of wave, and again becomes upright at the next trough ; it then reaches its greatest angle from the vertical on the opposite side, and finally is upright again at the next crest. The normal thus passes through a complete phase of motion in a time equal to the period of the wave. The character of this movement is the same as that of a vessel when performing a double roll in still water ; and if the double period of the ship coincides with the period of the wave, the motions of each *synchronise*, or keep time, with the other. Thus, the first deduction from theory is that if such synchronism occurs the vessel will speedily capsize. This is analogous to the case of a simple pendulum, which can be made to have considerable oscillatory motion by successive small impulses, if they are carefully timed with reference to the motion of the pendulum.

On comparing the above with the results obtained from the behaviour of ships at sea, it is seen that this theory correctly predicts the *character* of the motion, but not its actual amount. For example, if a ship falls in with waves of synchronising period, although she does not capsize under their action, her rolling will then be the heaviest she is likely to experience, *resistance alone preventing her from overturning.* This operates to limit the angle of roll in the following manner. When the waves first begin to act upon the ship, the rolling is only of moderate amount, and therefore the resistance is not great; the energy given by the wave to the vessel as it passes her is thus greater than that taken from her by the action of resistance, and the rolling is consequently increased. But this increase also augments the resistance, until finally the energy imparted by the wave is equalled by that abstracted from the ship by the resistances; and when this balance is attained, no further increment of roll will take place. Resistance also operates in a similarly beneficial manner in all other cases.

The effect of synchronism was well shown by the behaviour of the *Monarch* during the cruise of the Combined Squadrons in 1871. This ship is usually remarkably steady, but on one occasion, amongst waves having a period about twice that of her own, she rolled 26° from the perpendicular, whereas the *Prince Consort*, a noted heavy roller, only registered at the same time 15°. A similar event occurred during the cruise of a French squadron in 1863. As a general result, the *Solferino* proved by far the steadiest vessel in the squadron; but once, when subjected to a long low swell abeam, she rolled more heavily than the others, owing, undoubtedly, to approximate synchronism between the ship and waves.

As an instance indicating the correctness of theory in concluding that a ship of quick period will keep her deck approximately parallel to the wave, reference may be made to the rolling of the American monitor *Miantonomoh.* This vessel has a metacentric height of 14 or 15 feet, conducing to a very quick period (about $2\frac{3}{4}$ seconds); and as the reports of her performance state that but little water was shipped on her upper deck—situated from $2\frac{1}{2}$ to 3 feet above water—when crossing the Atlantic broadside on to large waves, it is clear that her deck must have kept nearly parallel to the surface of those waves. An extreme example of this is a raft floating amongst waves; its motion is so rapid relatively to that of the wave, that the mast continuously points along the normal to the wave as it passes, thus taking up the successive positions shown in Fig. 41. Such a raft-

FIG. 41.



like vessel would not be a steady gun platform, as, during the passage of about one-half the length of the wave, it would roll from an angle equal to the maximum slope of the wave on one side of the vertical, to the same angle on the opposite side; this arc of oscillation, amongst ordinary Atlantic storm waves, would be about 18°. The same unsteadiness of gun platform is also exhibited by actual quick-moving vessels, although one exception must be noted. When the short period is due to the adoption of great beam, such a vessel is far from satisfying the condition mentioned in the beginning of this chapter, of smallness in relation to the wave; and thus the assumption that the fluid pressure acts normally to the wave surface becomes vitiated. This pressure is much more nearly vertical than given by the above hypothesis, so that broadness of beam is equivalent to a reduction in the steepness of the wave, and such vessels may be steady. For instance, the Russian yacht *Opit* (late *Livadia*), with her 153 feet beam and 235 feet length, only rolled 4° and pitched 5° from the vertical when encountering very heavy weather in the Bay of Biscay.

Finally, many examples may be cited to show that a long period leads to steadiness. As one instance it may be mentioned that an analysis of the records taken of the rolling of the *Prince Consort* for a period of one day, when with the Channel Squadron in 1865, gave 37° as her mean arc of oscillation, whereas the *Achilles*, a vessel of longer period, recorded on the same occasion and under similar circumstances only 8° of roll; and this is but typical of the general relative behaviour of the two vessels throughout the cruise. Other examples will appear later on.

Since both theory and observation agree that synchronism between the ship and waves leads to heavy rolling, it is evident that this should be avoided if possible. It has been pointed out that for large angles of roll a vessel has a still-water period differing from that for small arcs, and thus, if at first synchronism occurred between the ship and wave, it would be destroyed when the vessel rolled deeply, and its effects thereby mitigated. But as the ship must obviously acquire considerable rolling motion before non-isochronism is possible, other means must be employed to attain the same end. This can be done by giving a still-water period to the ship greater than the half-period of any waves she is likely to encounter, thus making synchronism impossible when she is lying broadside on, save only in exceptional circumstances. The largest ordinary Atlantic storm waves have been found by observation to have a period of 10 or 11 seconds, so that if a ship is so designed as to have a still-water period greater than 5½ seconds, she will avoid synchronism when in the trough of all waves up to and including the largest ones ordinarily met with in the Atlantic, and will thus prove a steady gun platform at sea. As has been previously mentioned, little control is possible over the radius of gyration of a ship of given principal dimensions, and hence the period is made long by diminishing the metacentric height as much as possible, bearing in mind the necessity for securing sufficient stiffness. That smallness of metacentric height conduces to good behaviour has been abundantly proved by experience. For example, when the French vessel *Gloire* was found to roll heavily, the weights of the sister ship *Normandie* were lowered, owing to the mistaken view being entertained that "top-heaviness" was the cause of the excessive rolling. This increase of metacentric height, however, only caused her to roll more quickly and deeply than the *Gloire*, and this led to the raising of the weights, a proceeding which had a beneficial effect on her performance. Again, the *Prince Consort* has a metacentric height of 6 feet, leading to a period of about 5½ seconds, or that period most conducive to heavy rolling amongst ordinary Atlantic storm waves. The *Achilles* has a metacentric height of only 3 feet, this difference accounting for the longer period of the latter, and for the different behaviour of these vessels amongst waves, adverted to above. As another illustration, it was mentioned in a report on the *Imperieuse*, that her roll was only 3¼° at a time when the *Colossus* registered 17°, a result which must have been largely due to the smaller metacentric height of the former vessel.

Synchronism may also be avoided by altering the course and speed of the ship relatively to the waves. If this equality between the period of the ship and the half-period of waves occurs when she is broadside on, it may be destroyed by steaming obliquely towards, or away from, the waves. In the former case the *apparent* period would be diminished, and in the latter, increased ;

and as it is the apparent period which determines the time elapsing between the action on the vessel of successive waves, it is this period which must be compared with that of the ship. On the other hand, altering the course may bring about synchronism, and lead to heavy rolling. This was clearly demonstrated during some trials with the *Devastation*. When lying broadside on to waves having a period of about 11 seconds, her maximum observed roll to windward was $6\frac{1}{2}°$, and to leeward $7\frac{1}{2}°$. When she steamed obliquely from the waves so that their apparent period became $13\frac{1}{2}$ seconds, and twice that of the ship, the inclinations registered were 13° and $14\frac{1}{2}°$ respectively. Also, Mr. Froude records that when the *Active*, in carrying out some towing experiments, advanced obliquely towards a scarcely perceptible swell, little motion of the vessel was discernible; but in going obliquely from it, she rolled deeply, the apparent period of the wave having been lengthened into approximate synchronism with that of the ship.

As it is resistance alone that prevents a vessel from overturning in the critical case of synchronism, and as it lessens rolling in all other cases, it should be made as great as possible. The part most easily augmented is bilge keel resistance, and experiments have as conclusively proved its value in this connection as for speedily bringing a vessel to rest when rolling in still water. One of these, conducted by Mr. Froude, consisted in instituting a comparison between the rolling of the very similar vessels, *Greyhound* and *Perseus*, when subjected to the action of the same series of waves. They were each so weighted as to have an identical still-water period, and the only difference between them lay in the fact that whilst the latter had no bilge keels, one was fitted on each side of the former, 100 feet long, and $3\frac{1}{2}$ feet deep. They were taken outside Plymouth Breakwater and tried on three different occasions, the result in each case being that the *Greyhound* only rolled one-half as much as the *Perseus*. For example, the mean of twenty successive rolls, recorded at one of the trials, gave 6° for the former, with a maximum roll of 7°; the corresponding figures for the latter were 11° and 16° respectively. The model of the *Devastation* was also tried amongst waves, with bilge keels of different depths. With a depth of bilge keel on each side corresponding to 3 feet in the actual ship, the maximum angle to which the model was inclined by the waves was $13\frac{1}{2}°$; but when those keels were replaced by others 6 feet deep, the greatest inclination attained when encountering similar waves was only 5°. Experience, again, has firmly established the utility of bilge keels. It was stated before the Committee on Designs for Ships of War, in 1871, that previous to the fitting of bilge keels to the Indian troopship *Serapis*, a heavy Atlantic beam sea would cause her to roll considerably over 30°, whereas after they were fitted, the maximum roll under similar circumstances was only 23°. Bilge keels are therefore fitted to all H.M. Ships, and to many vessels in the Merchant Service.

A steady gun-platform can be secured, as previously stated, by the adoption of a small metacentric height. But in some ships a great metacentric height is unavoidable, as, *e.g.* in the *Inflexible*, where a large value of G M is necessary to provide against possible damage (*see* page 23). Such vessels will have quick still-water periods, and are therefore likely to prove heavy rollers. To remedy this, *balance chambers* have been fitted; they are simply tanks, fitted to contain water, and extend transversely across the ship. When the chambers are partially filled, the water moves from side to side as the vessel oscillates, and its effect is to add considerably to the resistance to rolling. Owing to the action of gravity, the water continually tends to run *downwards*, and thus, roughly speaking, it may be said to move in an opposite manner to the men employed to create rolling in a ship by running from side to side. Hence, as the movement of the men *augments* rolling, that of the water *retards* it. Experience with the *Inflexible* was conclusive as to the value of the chamber fitted in that vessel above the armoured deck aft. With $8\frac{1}{4}$ feet metacentric height, her period was only $5\frac{1}{2}$ seconds, and yet, whilst encountering waves in the Bay of Biscay having an apparent period nearly double her own, the maximum inclination from the vertical was less than 11°.

At very small angles of roll, the principal transference of water takes place in the form of a wave; and when its period in going from side to side is equal to the period of the ship, experience and theory show that the contained water then has its maximum effect in extinguishing the oscillations. Now, the period of this wave, in a chamber of given dimensions, is dependent upon the depth of water, and thus, although at all depths the water exercises a beneficial effect, there is one particular depth which will give the best result, and which can only be determined by experiment.

A careful series of experiments, with a model of the balance chamber of the *Edinburgh* fitted in an appropriately weighted oscillating frame to represent the ship, as well as trials upon the vessel herself, showed that from the results of model experiments, the effect of the water on the full-sized ship could be very reliably ascertained. Model experiments may therefore be substituted for those on the ship, thus entailing less expense and enabling results to be obtained for larger angles than would be possible in the full-sized ship, as well as affording the means of determining the best size of chamber and the quantity of water necessary to obtain the maximum effect. These trials gave 100 tons as the most effective quantity of water for the one chamber fitted in the *Edinburgh*, and also proved that its effect at small angles in reducing rolling was much greater than could be obtained by adopting any practicable depth of bilge keels. Its value at various angles as compared with the additional resistance which would be obtained, at those angles, by adding two feet to the depth of the bilge keels already existing on the ship, is given in the table below.

At 4°	the balance chamber adds	6 times	as much as two
„ 6°	„ „ „	„ 3 „	feet additional
„ 12°	„ „ „	„ the same	depth of bilge
„ 18°	„ „ „	„ one-half	keel.

This efficiency at small angles is important, as it is only in moderate weather that naval actions are likely to be fought, when steadiness of gun platform is of primary importance.

Before leaving this part of the subject it may be mentioned that although, as stated in Chapter II., free water leads to a reduction of stability, the diminution due to the use of the very moderate amount of water in a balance chamber can be safely neglected in the large stiff ships in which the device is employed.

The theoretical case of *unresisted* rolling amongst waves assumes, of course, no sail spread; and, as previously remarked, this investigation only points out *tendencies*, and not the actual *amount* of roll when resistance is included. But by the most valuable process known as *Graphic Integration*, the effects of fluid resistance, and air pressure on sails, may be taken account of, and by its means the Naval Architect is enabled to predict with considerable accuracy the behaviour of a ship (of which the still-water period and curve of extinction have been ascertained) when amongst any given waves, whether regular or not. For a general description of this method the reader must be referred to the *Manual of Naval Architecture* already referred to, wherein also will be found described the effect of sail spread upon the safety of a vessel. It need only be remarked here that since a vessel carrying sails is exposed to the action of gusts and to the steady action of the wind, in addition to the forces acting upon a mastless ship, a larger range and area are necessary for the curve of stability of the former than for that of the latter.

To ensure that the rolling observations conducted on board H.M. Ships shall be as useful as possible, every effort should be made to register simultaneously the periods and dimensions of the waves, and also the course and speed relatively to the waves, since these elements so largely affect the behaviour of the vessel. It is only by having a thorough knowledge of the surrounding circumstances that a comparison can be instituted between the results obtained at sea and those derived from theoretical considerations, a comparison which is necessary in order to further verify the accuracy of the theoretical method of procedure.

CHAPTER IV.

MATERIALS FOR SHIPBUILDING AND MODES OF CONNECTING.

Up to a period later than the middle of the present century, wood continued to be the material principally employed for shipbuilding purposes in the Royal Navy, as well as in the Mercantile Marine, although iron had been used on some occasions in both services before that time. The earliest notice of a vessel built of iron occurs in a publication of 1787, in which attention is drawn to a canal boat constructed of that material; but it was not until 1822 that an iron-built vessel—the *Aaron Manby*—proceeded to sea. Of the iron ships built for the Royal Navy before 1850, the *Simoom*, originally intended for a frigate but afterwards converted into a troop-ship, is probably the best known.

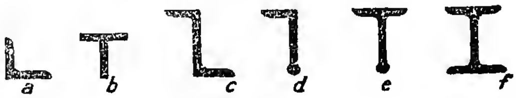
The destruction of the Turkish fleet at Sinope in the November of 1853, through the use of shells by the Russians, emphasised the fact that wooden vessels were exposed to the incendiary action of such shells; to prevent which, as well as to protect the guns and gunners, the French adopted iron armour. England commenced the use of armour a little later, but, unlike France, also adopted iron at the same time for the construction of the *hulls* of her war vessels. The only exceptions were some wooden ships which were partly built at the time armour was definitively adopted, and which, when afterwards armoured, produced the "converted" ships *Repulse*, *Prince Consort*, and five others; and a few more such as the *Lord Warden* and *Lord Clyde*, which were built of wood in order to usefully employ the timber stored up in the dockyards. About the same time iron began to be widely used in the Merchant Service, so that wood, as a material for shipbuilding, has now almost entirely disappeared in favour of iron or its newer competitor steel.

The adoption of iron, besides greatly reducing the risk from fire, led to other important advantages, not the least of which was the attainment of *lightness combined with strength*. In the best wood-built war ships, fully one-half the total weight of displacement was absorbed by the weight of hull, thus leaving only about one-half that displacement available for machinery, guns, armour, &c.; but with iron a much stronger hull may be given to the largest armour-clads than would be possible with wood, and with a weight of material of from 30 to 40 per cent. of the displacement only. The weight so saved from the hull can be applied to increasing the powers of offence or defence of a ship of war, by carrying heavier guns, thickening the armour, &c.; whilst in a merchant vessel, where a similar saving results from the use of iron, it renders possible the carriage of more cargo with a given total displacement, thus augmenting her earning power. Not only so, but iron-built ships are *more durable* than those constructed of wood; and the parts liable to decay in the former are not important for structural strength and can be easily repaired or replaced, whereas in the latter the reverse is true, and the necessary repairs are expensive. Iron ships, when properly designed and constructed, are also much *safer* than wood-built vessels.

Before passing on, reference must be made to one rather serious disadvantage attending the substitution of iron for wood. Fouling of the bottom due to the adhesion of seaweed, shell fish, &c.—to which all ships are liable, and particularly those employed in tropical waters,—can be easily prevented in wooden vessels by nailing sheets of copper on their bottoms; soluble salts of copper are formed by the action of the sea water, and these are slowly washed or dissolved away, carrying with them the substances attaching themselves to the surface of the copper, and so keeping that surface clean. This gradual wasting away of the copper is called *exfoliation*. In iron-built vessels, however, fouling cannot be prevented in the same simple way; for if copper sheets were attached directly to the iron bottom, galvanic action would ensue, and would lead to the rapid wasting away of the iron hull. Thus it is that iron ships must either have their iron skins coated with some antifouling composition, which requires renewing at least once a year to prevent serious fouling and the consequent loss of maximum speed, or they must be first sheathed with wood and then coppered, an expensive proceeding not unattended with danger, as will be pointed out in Chapter VIII. The thin bottoms or iron ships are also more easily penetrated by rocks, &c., than the thicker ones of wooden vessels, but the adoption of efficient watertight subdivision, and especially the use of an inner bottom, reduces this disadvantage to very small dimensions.

The wrought iron used in the Royal Dockyards was strong, very reliable in character, and could be manufactured into a variety of shapes eminently suitable for shipbuilding. Besides its capability for being rolled into sheets or plates of various thicknesses, it could be made into long bars of different sections, the most important of which are shown, and their names given, in Fig. 42.

FIG. 42.

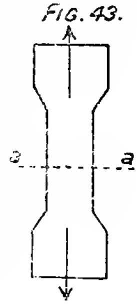


References.

a.	Angle-bar.	*d.*	Angle-bulb.
b.	T-bar.	*e.*	T-bulb.
c.	Z-bar.	*f.*	H-bar.

Such bars are known by the general name of *moulded* iron. To ascertain its fitness for service, this material had to undergo certain critical tests known as the *hammering*, *tensile* and *forge* tests. The first was conducted by striking each plate or bar with a hammer and searching for injurious surface defects ; the others were performed on specimen plates and bars only. The tensile test, instituted to determine the strength of the material, was carried out on pieces cut from a test plate both in the direction of its length and at right angles to that direction (*with* and *across* the grain) ; they were shaped as shown in Fig. 43, and then subjected to a pulling stress in the directions indicated by the arrows. When made of the best quality iron, these pieces were required to stand a stress, before breaking, of 22 tons for every square inch of material in the section through *a a*, when the test piece was cut from the plate *with* the grain, and 18 tons when cut *across* the grain. In the case of moulded iron it is evident that only the test in the direction of the grain could be made. The forge tests were of two kinds, *hot* and *cold*, and were undertaken for the purpose of trying the ductility of the iron. Specimen pieces of plate were required to bend to certain specified angles when both hot and cold, the angle *with* being greater than that *across* the grain ; and moulded iron had to stand bending into various prescribed shapes.

FIG. 43.



Notwithstanding the excellence of wrought iron, attention was early directed to the stronger substance *steel* (iron combined with a small proportion of carbon), as by its use the naval architect could advance still further in the direction of lightening the hull, and so adding to the carrying power of the vessel. That produced in this country up to 1875, however, although it had a high tensile strength—from 30 to 50 tons per square inch—was not uniform in strength or ductility, and was liable to failure whilst being worked into the ship. For these reasons it was but sparingly used, and then mainly for internal work, or for special vessels, such as those designed for service in shallow rivers or for blockade running, where lightness was of great importance. About the date just mentioned *mild steel*—containing less than one-half of one per cent. of carbon—was introduced, the employment of which is free from the disadvantages attending the use of the earlier steel. It has a lower percentage of carbon than that steel, which gives it greater ductility, although this is accompanied by a reduction in tensile strength ; it is still, however, much superior to wrought iron in this latter respect, as also in the former one. It is very little heavier than, and can be made into as various shapes as, wrought iron, and is so reliable and uniform in character that shipbuilders prefer using it to the last-named material. It has entirely displaced wrought iron for all important parts of the ships built for the Royal Navy, and has to a great extent done so also in those constructed for the Mercantile Marine. Each mild steel plate and bar is searched for surface defects, and *tensile* and *tempering* tests are applied to specimens. Strips cut lengthwise or crosswise from the test plates are required to stand a tensile stress of not less than 26 tons, nor greater than 30 tons per square inch, and must stretch at least 20 per cent. before breaking. This stretching (or *elongation*) is measured between two marks placed 8 inches apart on the specimens before the operation commences, so that after a test piece is broken and the parts laid together again, the distance between the two marks should be at least $9\frac{3}{5}$ inches. To still further prove the ductility of the material, the tempering test is instituted. This consists in heating strips cut lengthwise or crosswise to a low cherry red colour, and cooling them in water at a temperature of 82° F.; the specimens so treated must stand being bent double in a press to a curve of which the inner radius is equal to one and a half times the thickness of the plate tested. Moulded steel is also subjected to hot and cold forge tests to prove its fitness for service.

As an illustration of the saving in weight of hull accruing from the substitution of steel for iron, it may be mentioned that the weight of the hull of one of the earliest steel vessels, a fast unarmoured cruiser of 3738 tons displacement, was 1438 tons, whereas had iron been employed, the weight necessary would have been about 1613 tons. Supposing the saving to have been entirely devoted to increasing the coal supply, the component weights of the two vessels would compare as follows :—

	As built of steel. Tons.	If built of iron. Tons.
Armament and General Equipment ...	504	504
Machinery	1046	1046
Coals	750	575
Hull	1438	1613
Totals ...	3738	3738 tons.

That is to say, the employment of steel made it possible to carry 30 per cent. more coal than could have been carried by the iron ship without impairing her other qualities. Similar examples might be given for merchant vessels, Lloyd's rules allowing a general reduction of 20 per cent. in the thickness of all steel plates, &c., from what they would have been if made of iron. As, however, there are many internal iron fittings, woodwork, forgings, &c., not affected by this reduction, the total saving of weight is not so great a percentage as there indicated. Speaking generally, it may be said that the saving due to the use of steel amounts, in both war and merchant ships, to from 12 to 15 per cent. of the weight of iron which would be used in the iron ship.

Besides securing strength with lightness, other advantages are gained by the use of steel. In consequence of its superior ductility, many operations in the shipyard can be performed on it when cold which could only be done on iron after it had been heated ; and it is also less liable to crack when subjected to severe blows. That this latter quality is sometimes of importance may be seen from an example. The *Rotomahana*, a steel vessel employed in New Zealand, touched upon a rock, and a portion of her bottom plating was badly bent, *but no crack occurred.* A report on the accident states that the injuries were such that, had the bottom been of iron, so large a rent would inevitably have been made as would have caused her to founder in a few minutes. Another circumstance in favour of the employment of steel is its equality of strength and ductility lengthwise and crosswise. Iron, it will be remembered, has great differences in these respects in the two directions, and hence care was needed with that material in order to arrange it so that the greatest stress to which it would be liable should be taken in the direction of its length. No such precautions are necessary with steel.

If mild steel is to be worked hot, care should be taken not to do so when it is at any temperature between about 400° F. and 600° F., as at this "blue heat" the ductility is small, and cracking is likely to result. Very little difficulty, however, is experienced from this in practice, as the article is re-heated before attaining the above dangerous limit of temperature. If much work is done upon steel when hot, and it has to be re-heated, it should be subsequently *annealed.* This is carried out by heating the material in a furnace and afterwards allowing the fire to die out, thus permitting the steel to cool gradually whilst remaining in the furnace. This relieves the material from the condition of strain into which it had been thrown during working, and so restores its strength. Another necessary precaution when dealing with steel is to remove the "mill scale," which is formed on its surface during manufacture, from all plates liable to immersion in sea or bilge water. If this is not attended to, galvanic action will be set up in such water between the steel and adherent scale, which will result in the former being eaten away. To effect the removal of the scale, *pickling* is resorted to; that is, the plates are placed on edge for a few hours in a weak hydrochloric acid bath. After being taken out they have a stream of water directed upon them from a hose, their surfaces being brushed at the same time to remove the scale loosened by the action of the acid. Mill scale is also produced on iron plates, but its presence is not so injurious as in the case of steel, and no special steps are taken to remove it.

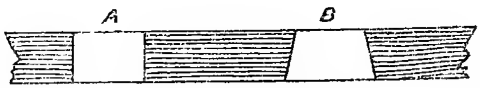
Cast steel is another material which is becoming extensively employed, taking the place of expensive wrought iron forgings and internal fittings of ships. It is stronger than that metal, cheaper, and the articles can be produced more rapidly—particularly where a number of the same pattern are required—since casting is a quicker operation than forging. This steel must stand a tensile test of at least 26 tons per square inch, with a minimum elongation of 10 per cent., besides having to undergo bending and other tests.

The above are the most widely used materials for shipbuilding; others which are employed will have their characteristics referred to as they are mentioned in succeeding chapters.

Instead of specifying the thickness of plates and moulded bars, it is usual to state the weight per square foot of material in the former case, and per running foot in the latter. For example, a plate weighing 20 lbs. per square foot, means a plate half an inch thick; one of 40 lbs., one inch thick, and so on in proportion: and an angle bar $3'' \times 3''$ of 7 lbs., means that each flange of the angle bar measures three inches, and that the bar is of such a thickness that a piece of it one foot long weighs 7 lbs. The thickness in this case is $\frac{3}{8}$-inch, and the dimensions of the bar may therefore also be expressed thus :—$3'' \times 3'' \times \frac{3}{8}''$.

A ship being made up of a great number of parts, the proper connection of these must be an important matter, as upon the efficiency of this connection depends the strength of the whole structure. Plates and bars are united by *rivets* of the same material as themselves, secured into holes made to receive them in the parts to be joined together. These holes are made either

FIG. 44.



by cutting out the material with a drill, or forcing it out by means of a punch ; in the former case the hole made is cylindrical, as shown at A, Fig. 44, and in the latter slightly tapered, as at B in the same figure, the smallest part of the whole being where the punch enters the plate.

The rivets are of two kinds, *through* (or clenched) and *tap;* the first pass right through the materials to be united, whilst the tap rivets usually do not, but effect a junction by screwing the pieces together. The through rivets are often made of a conical shape under the head as in Fig. 45, in order to fit into the holes made in punching. The points are formed on through rivets, by means of hammers, after the rivets have been heated and inserted into the holes intended to receive them, sufficient length having been left for this purpose. The shape of both heads and points depends upon the part of the ship in which the rivets are used, the forms ordinarily employed

FIG. 45.

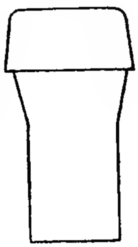
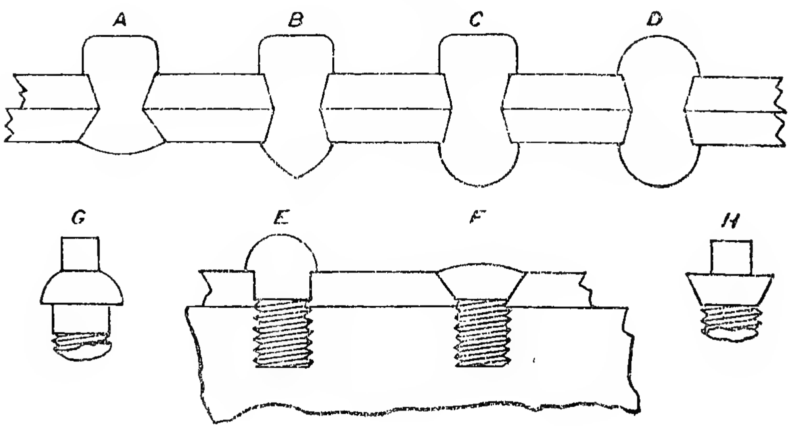


FIG. 46.

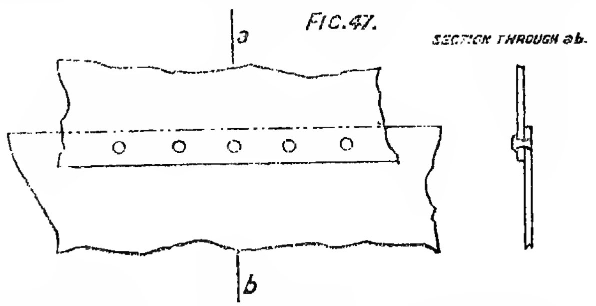


being shown in Fig. 46. That given at A has a *pan* head and *countersunk* point, the plating underneath being specially tapered away (or countersunk) to receive this point when beaten down. This form of rivet is used in those parts which are important for structural strength, experiments having proved such form to be very efficient. Pan heads and *conical* or *boiler* points (B, Fig. 46) are used for hand-riveted work not in sight, *snap* points, shown at C, being adopted in work which can be seen on account of their more finished appearance. Snap points are first beaten down roughly with hammers, and then finished off with a "snap punch"—a tool provided with a hemispherical hollow at one end—which is placed over the point and beaten until the point becomes rounded. For machine-riveted work, and occasionally for that put together by hand, snap heads and points, as at D, are employed. E and F show two forms of *tap* rivet; these are used either to connect a thin plate with a much thicker one, or in places where it would be impossible to knock down the point of a through rivet. The square studs shown on the tap rivets at G and H are necessary for screwing them into place, after which they are cut off, as indicated at E and F.

The operation of *punching* iron, and more particularly steel, injures the material in the neighbourhood of the hole, decreasing its strength. In the case of steel where the holes have afterwards to be countersunk, the injured material is removed by that proceeding and the remainder retains its normal strength. Or, where a countersink is not required, the strength of steel may be restored by the process of annealing previously described. If the holes are *drilled*, the strength of the material suffers no deterioration.

When one plate laps over and is joined to another, they are said to be *lap jointed;* such a joint is indicated in elevation and section

FIG. 47.



in Fig. 47. When, as in Fig. 48, they are connected by a third plate, the connecting piece is called a *butt strap* or *edge strip* according as it joins the ends ("butts") or sides of the plates.

FIG. 48.

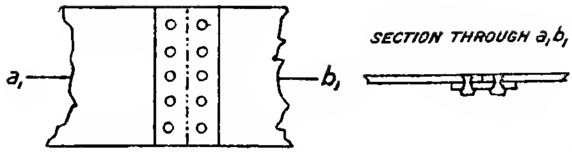
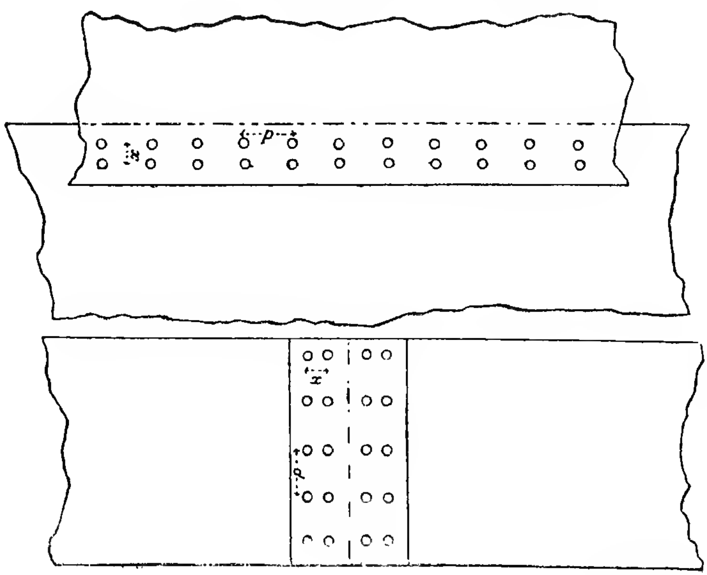


Fig. 48 shows the employment of a *single* butt strap ; but if two connecting pieces are used, one on each side of the plates to be joined, these plates are said to be *double* butt strapped. Angle bars are joined together by a covering strap of the same section, as shown in Fig. 48*a*.

FIG. 48a.

The size, number, and arrangement of rivets are all important matters when considering the fastenings of a joint. The size varies according to the thickness of the plates to be connected ; when these are thin, the diameter of the rivet is about twice their thickness, whereas for the thicker plates the diameter is not much greater than the thickness of these plates. For plates half an inch thick—a size much used in shipbuilding —the rivets employed are three-quarters or seven-eights of an inch in diameter. The number of rivets used depends upon considerations of strength, and upon whether the joint is afterwards to be made watertight or not ; if a watertight connection is required the rivets must be put closer together than would be otherwise necessary (*see* page 77). The arrangement of rivets usually adopted is that known as *chain riveting*, the rivets being placed in parallel rows with corresponding rivets opposite each other. Fig. 49 represents *double* chain riveting in

FIG. 49.



lap- and butt-joints respectively, *single* riveting being shown in Figs. 47 and 48. In *treble* chain-riveted butts there are three rows of rivets on each side of the joint. The distance, x, between the rows of rivets from centre to centre is $2\frac{1}{2}$ times the diameter of the rivets used ; and the distance between any two rivets in the same row, denoted by p in Fig. 49, is known as their *pitch*.

A joint is made watertight by *caulking* it. For lap-work, a tool called a *splitter* is employed to make a split in one of the edges,

FIG. 50.

and then another tool (a *maker*) is used to drive the piece so split tightly against the adjoining plate as at a, Fig. 50, thus making the junction watertight. In butt-work, a split is made on each side of the joint and, after the split pieces have been forced together the joint is rounded off with a hollow *reeding* tool, to the shape shown at b, Fig. 50. When an angle bar is to be caulked, the edges of its flanges are first chipped or planed square, so that the operation may be more efficiently performed.

Work to be afterwards caulked has its rivets closely spaced to draw the surfaces well together, and to ensure that they shall not spring apart under the operation of caulking. In such work the pitch varies from about 4 to 6 diameters of the rivet employed whereas in work not intended to be watertight the rivets are 7 to 9 diameters apart.

CHAPTER V.

CLASSIFICATION OF SHIPS.

Vessels may be divided into two great classes, *Armoured* and *Unarmoured.* The former comprises all those which are protected against gun attack by thick vertical armour plates, whilst the latter includes all from which this method of protection is absent. These two classes may be subdivided into others, according to the service upon which they are to be employed, the motive power used, their displacement, etc.; but it will be sufficient for the present purpose to draw special attention to that particular division into classes which depends upon the system of constructing the hull, leaving to Chapter XIV. their classification according to the distribution of armour and armament.

All modern war vessels belong to one of the three following classes :—(1) Iron or steel ships, (2) Iron or steel vessels sheathed with wood, (3) Composite ships. The first contains those in which all the structural parts are of iron or steel, including, amongst many others, the first class battle ships. The second division, as the name implies, consists of those iron or steel ships which have their skins afterwards covered with wood, with a view to being coppered. This method of construction is adopted in those vessels of the cruiser class which are provided with fairly large engine power and which are intended for foreign service, it being necessary that such ships should be able to keep the sea for considerable periods without their under-water surfaces becoming encrusted with shell fish, etc. If such fouling occurred, it would either necessitate docking and cleaning the ships abroad where accommodation for this is limited, or would lead to an undesirable reduction in speed. The vessels of the composite class are built of iron or steel with the exception of their bottoms, which are of wood. By this system the lightness and strength of the former materials may be associated with the superior resistance to penetration and ease in preventing fouling which attend the employment of wood. It can, however, only be carried out in comparatively small vessels of moderate engine power, as they are not sufficiently rigid to withstand the vibration produced by powerful machinery. If the advantages of a wood bottom are desired in a large ship, the necessary rigidity is secured by first working an iron or steel skin, as in the sheathed class just mentioned.

Many ships have the materials of which their hulls are composed differently arranged from those of others in the same class, in order to meet the requirements of strength, etc.; and it would be a somewhat long task to notice all the combinations adopted. Care, however, has been taken, in selecting vessels for detailed description, to choose such as will illustrate as far as possible these various methods of combination. Those selected are shown in Plate II. Fig. 51 represents in side elevation (or *profile*) an armour-clad of the *Admiral* class. It is protected at the side in the vicinity of the water line by an armour belt A A, of 18 inches maximum thickness and 7½ feet deep, about 5½ feet of this being below water. Above the armour a protective deck 3 inches thick extends horizontally across the ship, (excepting a slight slope near the sides,) and a deck of the same thickness is worked at each end of the vessel. These latter slope upward from the sides, but are worked horizontally in the centre. The four heavy guns are carried on revolving turntables in two fixed armoured redoubts or *barbettes*, marked B, the maximum thickness of armour by which they are protected being 14 inches. A *communicating tube* of 12-inch armour extends to each barbette from the protective deck, so that the ammunition may be shielded by armour during the whole time of its passage from the magazine to the guns. The *conning tower* C, from which the ship can be worked in action by the commanding officer, is also protected by armour plates 12 inches thick. A spar deck is worked from barbette to barbette, between which and the upper deck is carried a battery of 6-inch and quick-firing guns. Of the platforms and flats below the lower deck, the most important are the watertight platforms shown in the figure, where also is indicated the extent to which an *inner bottom* is employed. This latter would prove an important element of safety if the outer bottom in its vicinity became damaged, and it encloses a space between itself and the outer bottom which can be utilised for water ballast.

The general arrangement of a sheathed corvette of the "C" class (*Comus*, *Constance*, etc.) is shown in Fig. 52, from which it will be seen that a protective deck 1½ inches thick is worked over the engine and boiler rooms; and above this, coal is stowed at the sides. These vessels have no inner bottom, but as the combined steel and wood renders the skin difficult of penetration by rocks or other hard pointed substances, a great objection to its omission is removed.

In Fig. 53 is given the profile of a composite sloop of the

Buzzard type. A $\frac{1}{2}$-inch watertight deck covers the engines and boilers, and coal is carried above it. Here, again, no inner bottom is fitted, the wooden skin opposing considerable resistance to penetration.

Each of the hulls of the preceding ships may be divided into the several parts mentioned below :—

(1) *Keel, stem, and sternpost,* which form the lower and end boundaries of the vessel, and together constitute, as it were, the *backbone* of the ship.

(2) *The framing,* designed chiefly to stiffen the bottom against water and other pressure. The principal portion forms what may be called the *ribs,* worked transversely, and *longitudinal framing* placed fore and aft ; and other important parts are the *beams,* which connect together the ribs on opposite sides of the ship and form supports for the various decks, and *pillars* which help to maintain the beams in position.

Both *longitudinal* and *transverse* frames are employed in every ship to a greater or less extent, the prominence given to each set depending upon the type of vessel as well as on its position in the ship. The frames in the more important set are worked *continuously*—that is, the plates and angle bars of which each is composed are connected by butt straps—whilst those in the set of lesser importance are worked *intercostally*—or in short lengths —between the continuous frames.

(3) *The bottom and side plating or planking,* the primary function of which is to keep water out of the interior of the vessel.

(4) *The deck plating or planking,* forming platforms upon which the various weights, such as guns, stores, etc., are carried.

(5) *Watertight partitions,* which divide the vessel into watertight compartments.

(6) The multitudinous *fittings,* of which those in connection with the pumping, drainage, ventilating, and steering arrangements, are amongst the most important.

In addition to the uses just enumerated, these parts all, with the exception of the last, contribute greatly to the strength of the structure. They will be described in detail in the succeeding chapters, especially as regards the first three items, where the principal differences of construction occur ; and afterwards will be added some general observations concerning the method of protecting vessels from the attack of guns, preserving them from deterioration, etc.

Fig. 51.

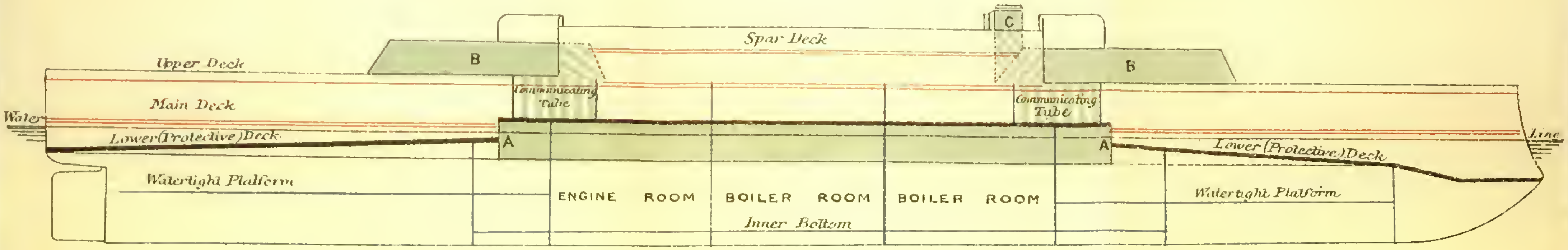


Fig. 52.

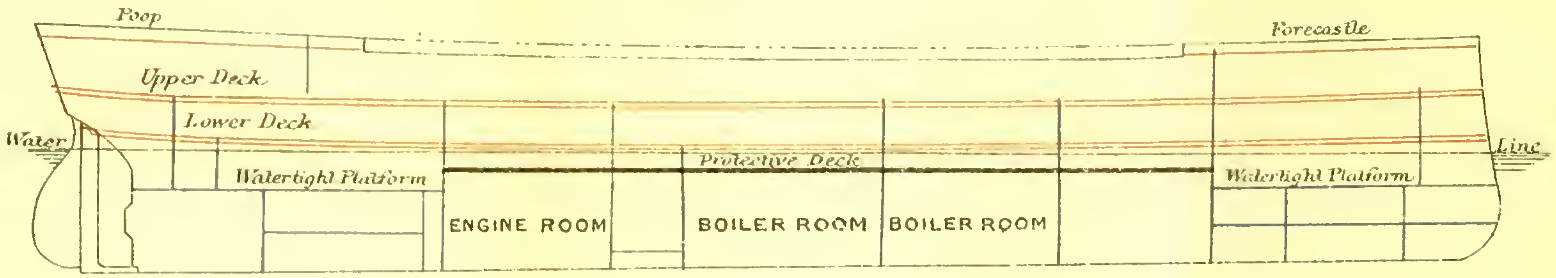
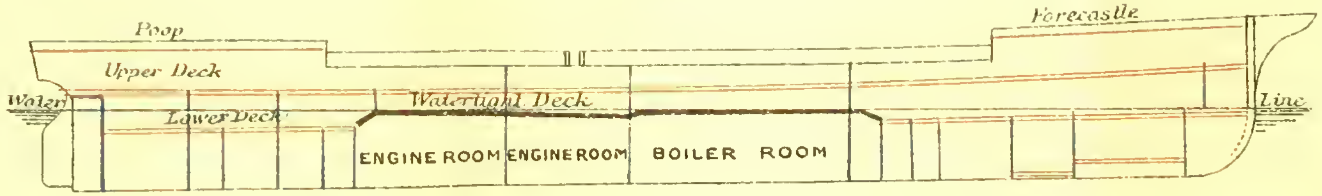


Fig. 53.



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CHAPTER VI.

KEELS AND FRAMING.

1. *Of an Armour-clad.*

FIG. 73 on Plate III. shows a half *midship section* of the vessel represented in Fig. 51 ; *i.e.*, a section which would be obtained by cutting the ship in the middle of its length by a vertical athwartship (or *transverse*) plane. This exhibits the general method of construction adopted for the length of ship over which the inner bottom extends, whilst Fig. 74, a transverse section taken beyond those limits, indicates the manner in which the ends are built. These sections show the position of the armour, the various decks, inner and outer bottom plating (the space included between these being known as the *double bottom space*), &c., besides the framing to be now specially described.

FIG. 54.

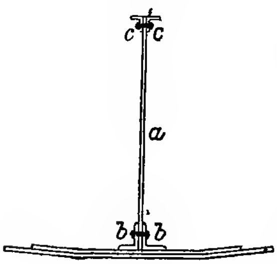
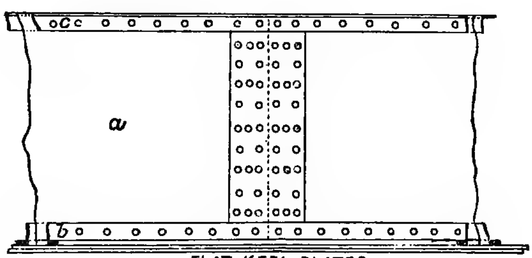


FIG. 55.



It will be seen from Plate III. that the outer bottom plating in the middle line of the ship is in two thicknesses; this plating forms the *flat keel*, upon which stands the *vertical keel* as shown on a larger scale at *a* in Fig. 54. Both keels extend from stem to sternpost, to which they are securely fastened (*see* Chapter VII.). The vertical keel is ½ inch thick, 38 inches deep throughout the length of inner bottom and a little less at the ends. It is connected to the flat keel plates by the angle bars *b*, *b*, the flanges of which are each 4 inches wide and ¾ inch thick; and to the inner bottom plating by the bars *c*, *c*, having flanges 3 inches wide and ⅜ inch thick. The several plates of which the vertical keel is made up are at least 12 feet long and are joined together by double butt straps, each rather more than $\frac{1}{4}$ inch thick, extending between the upper and lower keel angle bars. These straps are treble chain riveted, alternate rivets in the middle row on each side of the joint being sometimes omitted as in Fig. 55, where a short length of the vertical keel is shown in side elevation. The keel angle bars are worked in the longest lengths procurable, and their ends are connected by pieces of angle bar (*see* Fig. 48*a*) sufficiently long to take two rivets in each flange on each side of the joint. Throughout the length of the inner bottom, the butt straps and angle bars of the vertical keel are carefully *caulked*, thereby forming a watertight partition in the double bottom space. The inner flat keel plate is 39 inches wide and the outer one 48 inches, their thicknesses amidships being respectively $\frac{5}{8}$ths and $\frac{7}{8}$ths of an inch; at the ends they are slightly thinner, the combined thickness amounting to $1\frac{1}{8}$ inches. As these plates are an integral portion of the outer bottom plating, a description of their connections will be deferred until Chapter VIII.

The *armour shelf* upon which the armour rests, as shown in Fig. 73, is a continuous longitudinal frame (*see* page 80). It is 33 inches wide, $\frac{1}{2}$ inch thick, and extends beneath the armour throughout the length of the latter; it is then joined to, and forms part of, the protective deck forward and aft. The butts of the plates are connected by single butt straps $\frac{1}{2}$ inch thick and double chain riveted, the joints being caulked for watertightness. This shelf plate is united to the bottom plating by an angle bar $5'' \times 5'' \times \frac{1}{2}''$.

A reference to Fig. 73 will show that between the vertical keel and armour shelf there are four other longitudinal frames, or *longitudinals* as they are called, which stand at right angles to the bottom plating. These are known as the 1st, 2nd, 3rd, and 4th longitudinals from keel respectively, and are continuous throughout the length of the double bottom, whereas the *transverse* frames within the same limits are worked in short lengths from the vertical keel to armour shelf. On the contrary, Fig. 74 shows that the transverse frames towards the ends of the vessel are continuous from the vertical keel to the protective deck and from thence to the upper deck, whilst the longitudinal frames are mainly intercostal, a fact which is indicated by drawing these frames with *ticked* lines where they cross the continuous transverse frames.

Of the longitudinals in the double bottom, the third one from keel on each side of the ship is made watertight, so as to form watertight boundaries to the double bottom spaces included between themselves and the vertical keel. They are therefore unpierced, but the remainder have holes cut in convenient parts of their length, sufficiently large to allow of easy access from one side to the other when painting the compartments, &c.; this has the further advantage of decreasing the weight of hull. The angle bars on the inner and outer edges of each longitudinal, by which connection is made to the inner and outer bottom plating, are worked in long lengths joined together by covering pieces of angle bar; the bars on the inner edges are $3'' \times 3'' \times \frac{7}{16}''$, whilst the outer ones are $3'' \times 3\frac{1}{2}'' \times \frac{7}{16}''$, the $3\frac{1}{2}$ inch flange being that riveted to the bottom plating. The plates of the longitudinals are $\frac{7}{16}$ inch thick, and their widths are 35, 31, 30, and 30 inches respectively, commencing with the first from keel. These plates in each longitudinal are from 16 to 20 feet long, single butt straps, $\frac{1}{2}$ inch thick and treble chain riveted, uniting those of the watertight longitudinals, whereas double butt straps, each one-half the thickness of the longitudinal and double chain riveted, are employed for the others. The reason for the single strap in the former case is that the thinner plates of double butt straps could not be efficiently caulked; and treble riveting is adopted from considerations of strength. Side views of these connections are given in Figs. 56 and 57.

FIG. 56.

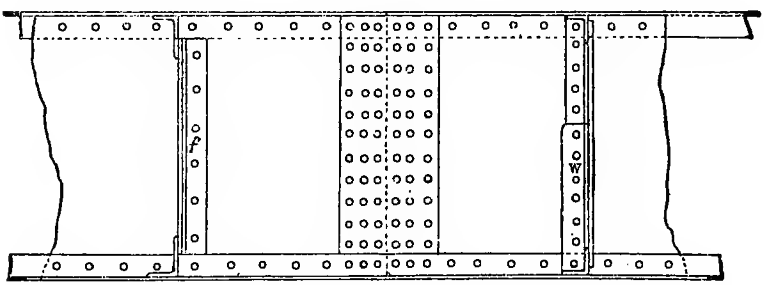
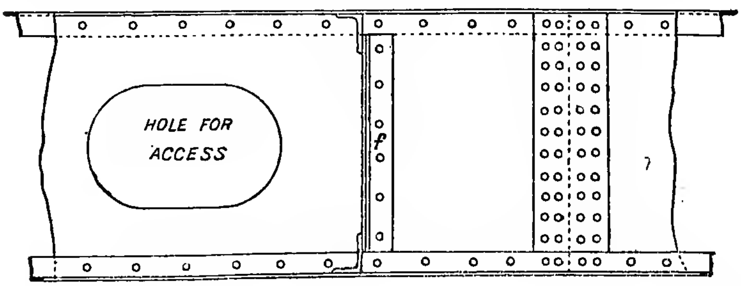


FIG. 57.



Before passing on, it may be mentioned that in the most recent armour clads, *two* of the longitudinals in the double bottom on each side of the ship are made watertight.

Throughout the length of ship for which an inner bottom is worked, the transverse frames which extend from the keel to the armour shelf are placed 4 feet apart, and consist principally of *bracket frames* (*see* Fig. 73, Plate III.) worked intercostally between the longitudinals. Fig. 58 shows on an enlarged scale a part of one of these frames, *viz.*, the length included between the vertical keel and the adjacent longitudinal. It consists of two *bracket plates* a, a, $\frac{5}{16}''$ thick; inner and outer angle bars b and c which secure those plates to the inner and outer bottom plating, and the sizes of which are $5'' \times 3'' \times \frac{1}{2}''$ and

FIG. 58.

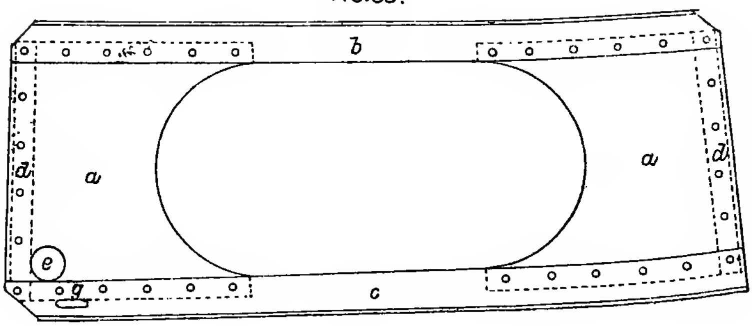
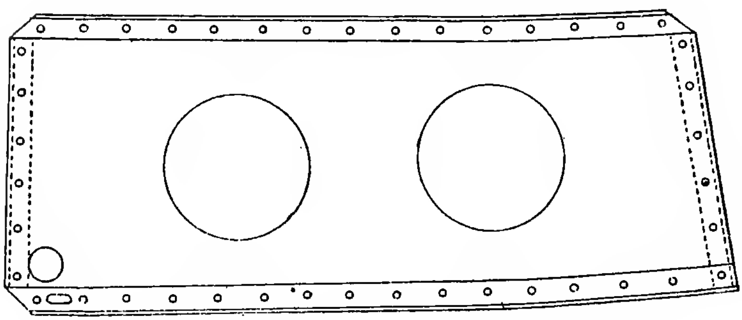


FIG. 59.

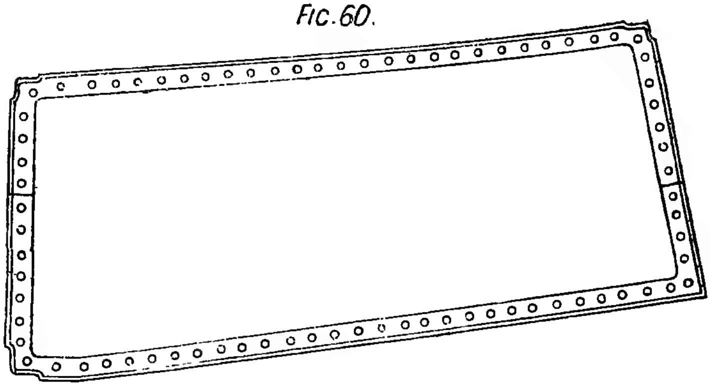


$5'' \times 3\frac{1}{2}'' \times \frac{1}{2}''$ respectively (the 5-inch flange in each case being vertical); and two small angle bars d, d, $3'' \times 3'' \times \frac{3}{8}''$, intended to connect the bracket plates to the vertical keel and longitudinal as indicated at *f* in Figs. 56 and 57. To avoid bending the inner and outer angle bars to fit over the keel and longitudinal bars, the ends of the former are cut off as shown. The small holes *e* and *g* are for drainage purposes.

Where, however, extra rigidity is required, as immediately under the armour or beneath the engines, solid plates lightened with holes are substituted for the bracket plates, the additional stiffness obtained by this substitution allowing smaller angle bars to be used. This method of construction will be noticed below the armour in Fig. 73 ; and Fig. 59 shows the kind of framing adopted under the engines, where the inner angle bar is $3'' \times 3'' \times \frac{3}{8}''$, and the outer one $3'' \times 3\frac{1}{2}'' \times \frac{7}{16}''$.

Finally, at intervals of 16 or 20 feet, *solid plate frames* are substituted for the bracket frames, except above the watertight longitudinal, where the ordinary construction is retained. A sketch of a part of one of these frames is given in Fig. 60, and its connection to a longitudinal is shown at W in Fig. 56. The inner and outer angle bars employed are of the same sizes as those just mentioned,

FIG. 60.



but they have their ends turned so that they may be connected to the longitudinals, the corners being shaped to fit closely against the angle bars of the latter so that the flanges of the frame angle bar may be efficiently caulked round them as well as against the plates of the longitudinals, bottom plating, and the plate of the frame itself, the rivets being closely spaced for that purpose. Of these watertight frames, one is fitted at each end of the double bottom space, and the remainder divide that space, up to the watertight longitudinal, into a number of watertight compartments.

The transverse frames behind armour, extending to the protective deck, are in two sets, and consist of long lengths of Z bar, $10'' \times 3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$; Fig. 61 shows a short piece of one of these bars, with the dimensions figured upon it. The frames of the first set are 4 feet apart, and extend down to the first longitudinal

Fig. 61.

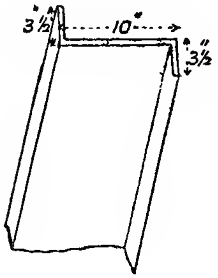
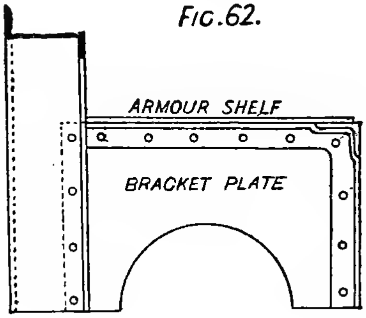


Fig. 62.



below the armour shelf as indicated in Fig. 73. From that figure it will be seen that each of the Z-bar frames laps against the plate of the corresponding transverse frame below armour, to which it is secured by a single row of rivets; and also that the outer flange of each frame is cut off from a point 2 feet below the armour shelf, as it is not required for structural purposes, and its removal decreases the weight of hull. The frames of the second—or *intermediate*—set are worked midway between those of the first, and there are therefore no frames below armour to which their heels may be attached. These heels only run down for a distance of 2 feet below the armour shelf, as in Fig. 62, and each is secured by a single row of rivets to a bracket plate $\frac{5}{16}''$ thick, which is connected to the armour shelf and bottom plating by a $3'' \times 3'' \times \frac{7}{16}''$ angle bar. The upper ends of the frames in each set are united to the protective deck by short pieces of angle bar.

FIG. 63.

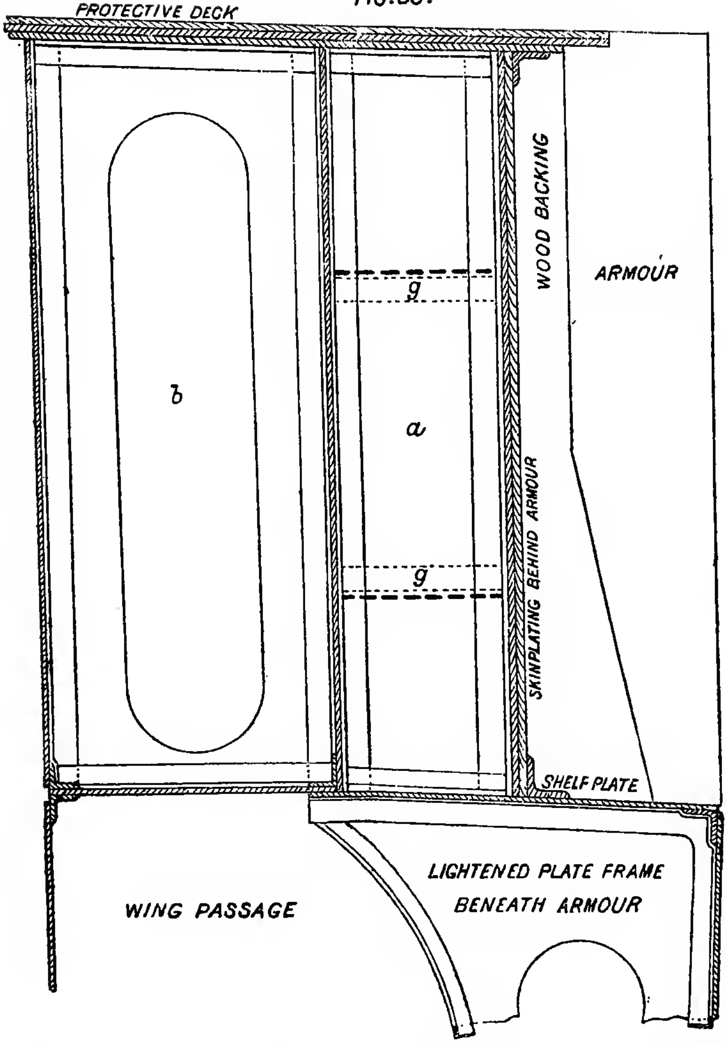
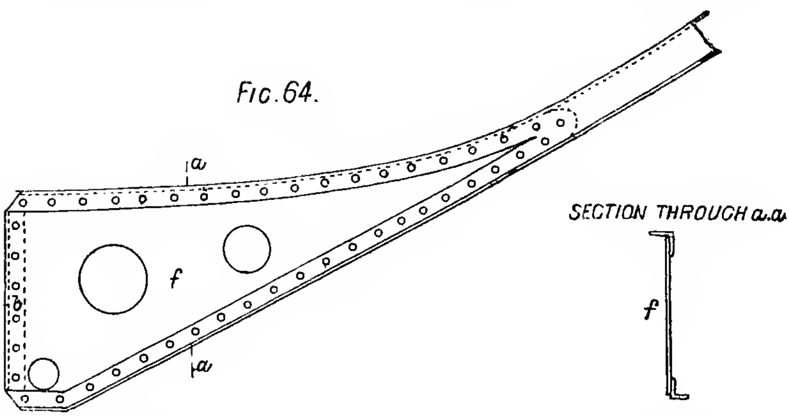


Fig. 73 shows that outside the plating secured to the frames just described, three *horizontal girders* marked *g* are worked, composed of plates and angle bars; these, crossing as they do the transverse frames at right angles, greatly stiffen the supporting structure behind armour.

It will be convenient to notice in this connection the important differences of construction introduced into this part of later armour clads. Experiments having shown the necessity for rigidly supporting the armour now used in order to bring out its maximum powers of resistance, the strong plate frames shown in Fig. 63 have been substituted for the weaker Z bar frames. There are two sets of frames, which extend from the protective deck to the armour shelf only. Those of the outer set—one of which is marked *a* in the figure—are 2 feet apart, each consisting of a solid plate 21″ wide and $\frac{5}{8}$″ thick, with a single angle bar on its outer edge and double angle bars on its inner edge, by means of which latter it it secured to a $\frac{1}{2}$″ plate. On the other side of this plate the frames of the second set are worked, 4 feet apart, each being made up of a solid plate from which the central part has been cut to the extent shown at *b* in order to provide a passage way at the back of the armour. Each frame is 3 feet wide and $\frac{1}{2}$″ thick, and has double angle bars on its outer edge and a single bar on the inner one. There are here two horizontal girders *g*, *g*, consisting of short pieces of plate and angle bar fitted between the first set of plate frames. The foregoing combination is known as the "box-backing" type of framing.

Returning now to the framing of the *Admiral* class, the transverse frames beyond the limits of the double bottom, both above and below the protective deck, are 3 feet apart, and, in the later vessels of that class, as well as in all succeeding armoured ships, are made of Z bars $6'' \times 3\frac{1}{2}'' \times 3'' \times \frac{3}{8}''$, the $3\frac{1}{2}''$ flange being the one secured to the bottom plating. As before pointed out, the part below the protective deck is worked continuously between that deck and the vertical keel; and the lower end of each frame is split for some distance along the middle of the web, thus producing two angle bars. These are opened out as in Fig. 64, and

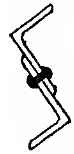
FIG. 64.



a *floor plate f*, $\frac{3}{8}$″ thick, is riveted to them, by which the strength of the frame is greatly increased. The heels of these frames are each connected to the vertical keel by a short piece of angle bar *b*, whilst the heels of those *above* the protective deck are secured thereto by means of bracket plates and angle bars as indicated in

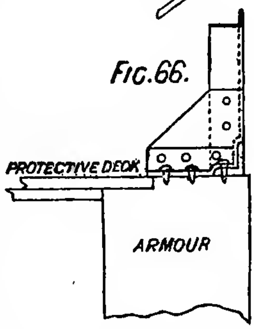
Fig 74. Instead of these Z bars, earlier vessels have the frames at their ends made up of two angle bars riveted back to back as shown in section in Fig. 65, the outer or *frame* angle bar being $5'' \times 3\frac{1}{2}'' \times \frac{7}{16}''$, and the inner or *reverse frame* bar $5'' \times 3'' \times \frac{7}{16}''$. Below the bilges they are opened out to receive a floor plate.

FIG. 65.



The remaining transverse frames between the protective and upper decks are spaced 2 feet apart, and are of two kinds, worked alternately. The first consists of angle bars $7'' \times 3'' \times \frac{3}{8}''$, having their lower ends turned as indicated immediately above the armour in the midship section on Plate III., and secured to the armour and protective deck by tap rivets; whereas the second are made of smaller angle bars, $4'' \times 3'' \times \frac{3}{8}''$, having their heels fastened to the armour by means of light bracket plates and angle bars, as illustrated in Fig. 66, their upper ends being similarly secured to the plating of the upper deck.

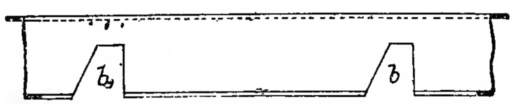
FIG. 66.



The nature of the framing between the upper and spar decks is determined by the fact that guns are fought there behind thin plating. It is therefore inadmissible to employ closely spaced transverse frames on account of the splintering which would ensue should the unarmoured side be struck by shot. Instead thereof, splinter-proof partitions or *traverse bulkheads*, 1″ thick and stiffened with vertical angle bars 2 feet apart, extend between the guns from the side of the ship to *a* (Fig. 73) so as to restrict any splintering caused by shot or shell to a limited area; and a horizontal Z bar $10'' \times 3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$ is worked between the traverse bulkheads to stiffen the side plating.

The longitudinal frames at the ends of the ship consist of Z bars of the size just mentioned, cut at intervals of 3 feet in the manner shown at *b* in Fig. 67, so that when put over the transverse frames from inside the ship, the outer flange may be connected to the bottom plating (*see* sections of these in Fig. 74). The part of each longitudinal which projects inside the transverse frames is riveted to a single

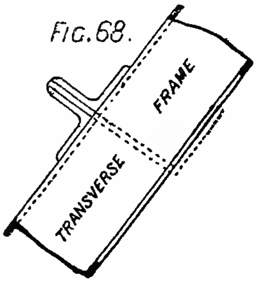
FIG. 67.



continuous angle bar $3'' \times 3'' \times \frac{5}{16}''$, this latter being also securely fastened to the inner flanges of the transverse frames. Near the keel, where the width of the athwartship frames precludes the use of Z bars, the longitudinals are made up of plates worked inter costally between the frames, each being secured to the outer bottom and to a continuous piece of plate riveted inside the frames, by short pieces of angle bar, as indicated in Fig. 74.

In some recent vessels the Z-bar longitudinals have not been worked, their places being taken by short lengths of angle bar which fit between the transverse frames and project sufficiently far inside them to take hold of two continuous angle bars ; a sketch of one of these longitudinals is given in section in Fig. 68.

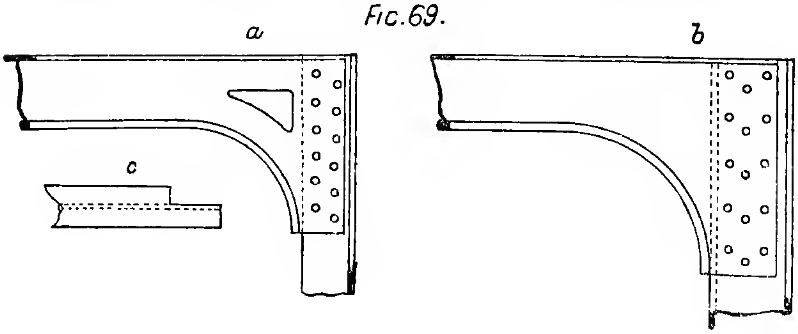
FIG. 68.



The above completes the description of the *main* frames of the ship; and attention will now be directed to the subordinate but important parts of the framing which stiffen the main frames, *viz.*, the *beams* and *pillars*.

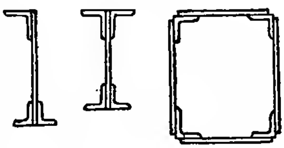
Beams are usually made of long bars of moulded iron or steel and one is worked from side to side under each deck and platform at every transverse frame, except where these in armour clads are only 2 feet apart, and then they are connected to alternate frames only. Those to the spar and upper decks of the vessels under consideration consist of tee-bulb bars 9″ deep, the width of the top flange being $5\frac{1}{2}''$; whilst the beams under the remaining decks and platforms are of angle bulb section of the following sizes :— Under the main and lower decks, 9″ deep and $3\frac{1}{2}''$ flange, and beneath the platforms 5″ deep with a $2\frac{1}{2}''$ flange. Each of these beams has a *knee* or *arm* formed at each end, and these are connected to the corresponding transverse frames by two or three rows of rivets as

FIG. 69.



at a and b in Fig. 69, where is exhibited the riveting of a beam arm to one of the frames above and behind armour respectively. Where tee-bulb bars are employed, a part of the upper flange must be cut away at the ends as shown in plan c, Fig. 69, so that the beam may fit closely against the frames. To save weight, some of the beam arms are lightened by holes, as shown in Plate III. From that Plate it will also be seen that under the spar deck, where no transverse frames are worked, the beams are united to the side plating through the medium of angle bars. In some ships, under concentrated weights such as turrets, &c., the beams are made of strong bars of H, or other similar solid, section; or they are built up with plates and angle bars, three usual sections being given in Fig. 70. Also the beams under platforms in hold are sometimes made of small Z or angle bars. A "round up" from the sides towards the middle line is usually given to the beams, in order that any water on the decks may flow readily into the gutterway. Those under that part of the upper deck between the barbettes are, however, not thus curved, the beams being kept level for convenience in laying the racers for the guns.

FIG. 70.



By supporting a beam in the middle of its length, its strength is considerably increased, and such support is given to each beam in the ship as far as possible, either by utilising the ordinary structural arrangements or by the use of *pillars*. An example of the former method may be seen in Fig. 73, where the divisional bulkhead between the engine and boiler rooms on opposite sides of the ship efficiently keeps the beams of the main deck in position at the middle line. Most of the pillars used are made tubular, as that form possesses greater lateral rigidity than solid ones having the same weight of material. They have their heads and heels welded in solid, and formed so as to fit against the beams or plating of the decks between which they are worked. Fig. 71 shows in front and side elevation one of the pillars under the

FIG. 71.

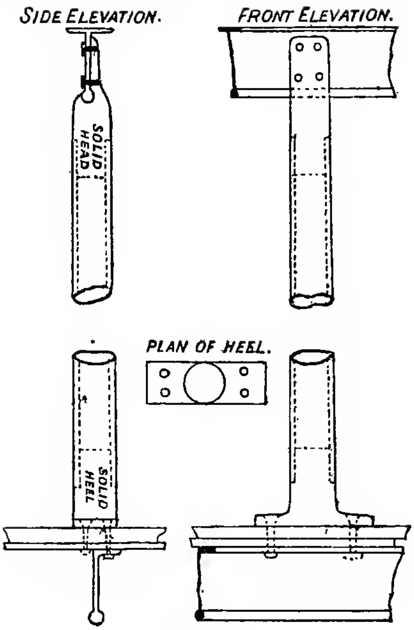
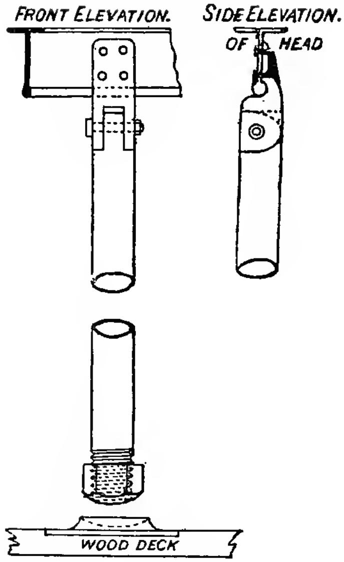


FIG. 72.



upper deck amidships; these illustrate the most usual shapes of both the heads and heels of pillars, and likewise show their fastenings. When a pillar is to be placed under a beam of H or other similar section, its head is formed similarly to the heel of the pillar just referred to.

In the vicinity of capstans, the pillars must be made capable of being moved out of the way when the capstan bars are to be used. Elevations of one of these portable pillars are given in Fig. 72, from which it will be seen that it is made to pivot about a bolt in its upper part, whilst its lower end carries a brass nut with a rounded under surface, working upon a screw thread; which nut, when the pillar is in place, is turned round by means of a spanner until it bears tightly against a brass casting of corresponding shape fixed to the deck plank. It is only necessary to turn back the nut so as to clear the metal casting, in order to hinge up the pillar whenever required; or, by removing the bolt securing the head, the pillar can be taken away if this is thought desirable. There are various other devices for securing the heels when the pillars are required to give support to the deck above, one of the most common being to drive them into wedge-shaped *shoes* fixed to the deck.

Fig. 73.

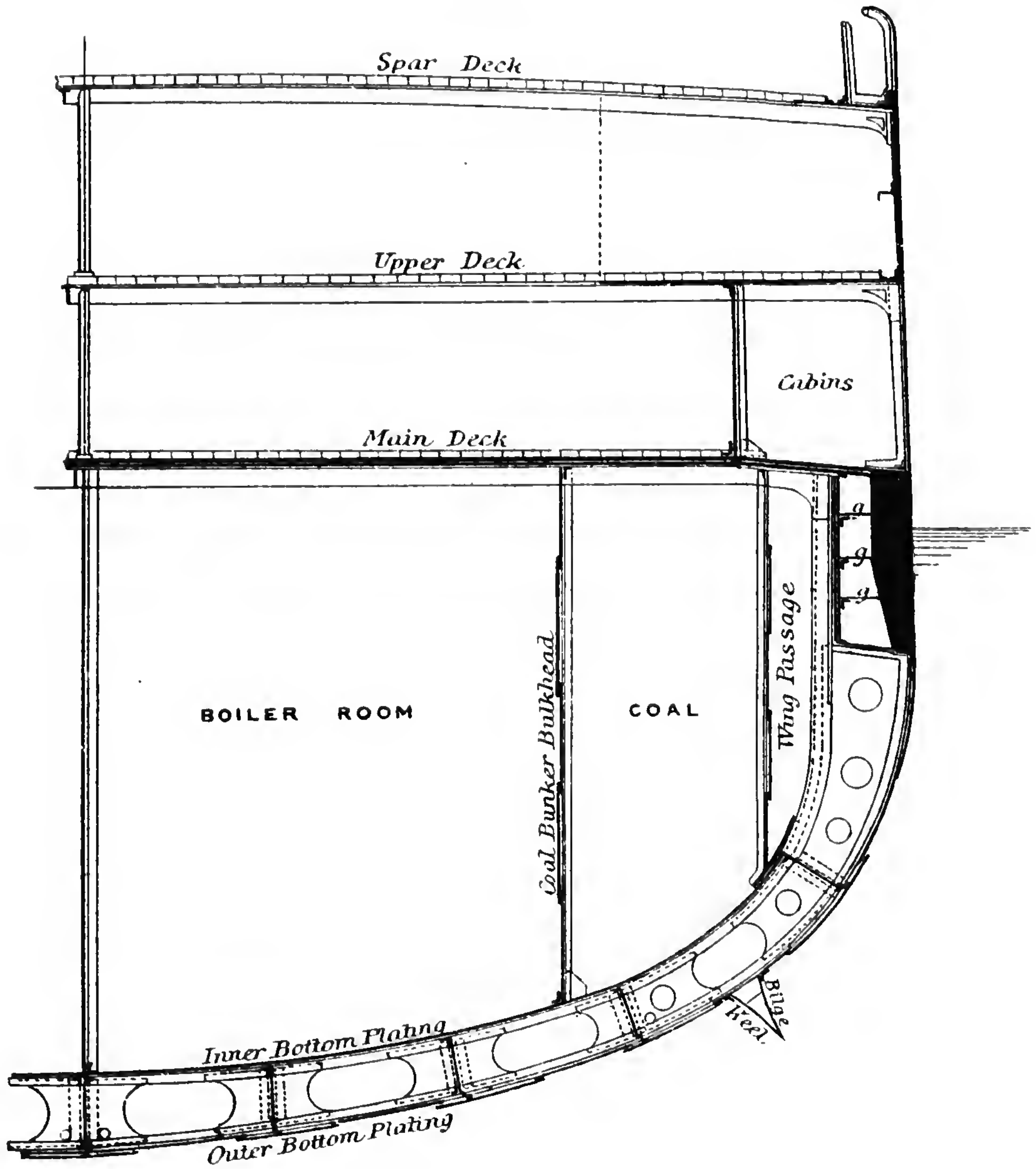
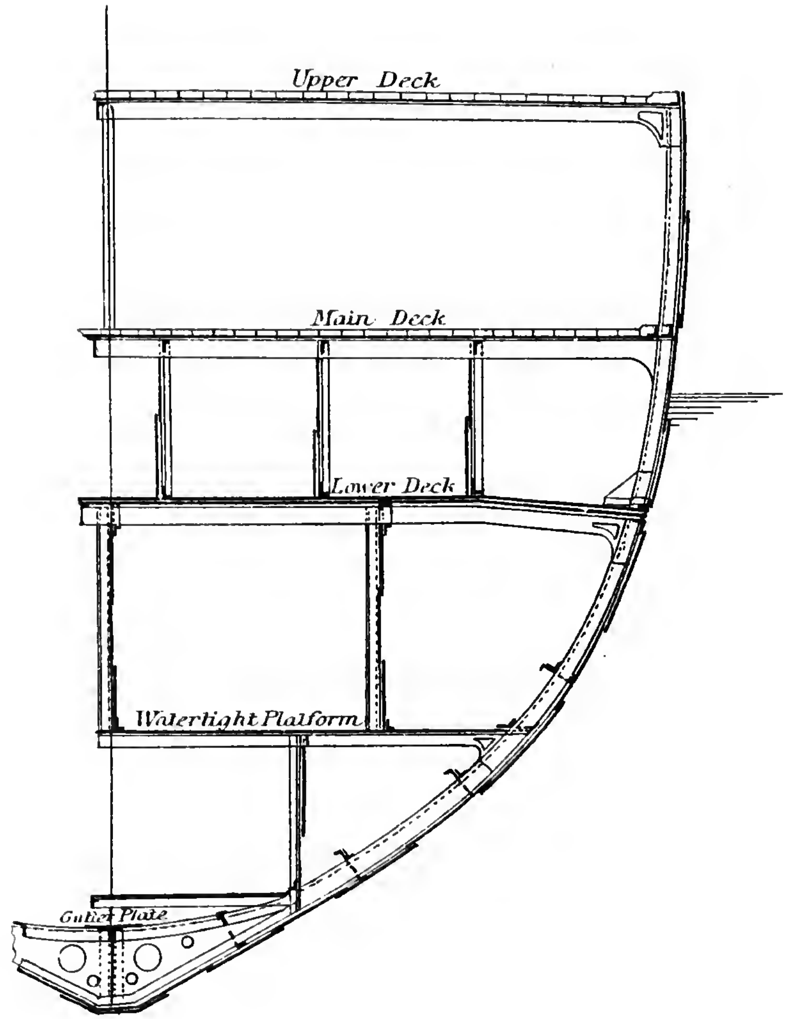


Fig. 74.



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The largest pillars employed in the above vessels are 7″ in diameter and $\frac{3}{8}$″ thick; and the smallest 5″ diameter and $\frac{1}{4}$″ thick.

2. *Of a Sheathed Corvette.*

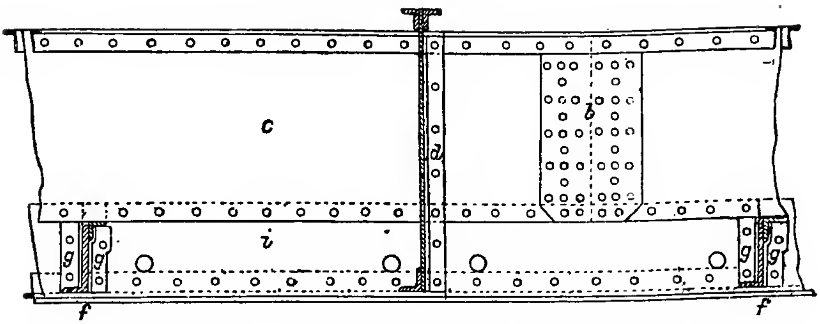
Fig. 76 on Plate IV. gives the midship section of the vessel shown in Fig. 52, whilst Fig. 77 represents a transverse section taken towards the ends. The framing, both longitudinal and transverse, is partly continuous and partly intercostal; and will be described in such order as—it is thought—will enable the method of construction to be best understood.

The flat keel consists of a single thickness of plate, $3\frac{1}{2}$ feet wide and $\frac{1}{2}$″ thick, extending continuously from stem to stern. Its connections, as also a description of the *wood* keel, will be given in Chapter VIII.

The transverse frames are $3\frac{1}{2}$ feet apart from centre to centre, those in wake of the protective deck being cut off thereby. Also, the framing above the deck differs from that below and beyond it. Every alternate frame of those below and beyond the protective deck extends continuously across the middle line of the ship, and is made up of a frame angle bar riveted to a reverse bar after the manner shown in Fig. 65; the frame angle bars are $8'' \times 3'' \times \frac{3}{8}''$ for the frames beneath the protective deck, and $6'' \times 3'' \times \frac{3}{8}''$ for those beyond it, whilst the reverse bars are $2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}''$ throughout. In Fig. 78 is indicated the lower part of one of these frames; and the upper ends of those below the protective deck are formed into a knee for security thereto in a manner similar to that shown at *a*, Fig. 76, whereas towards the ends of the ship these frames extend continuously from topside to topside. When it is not possible to make the frame or reverse frame of a single length of angle bar, the several pieces are connected together by covering angle straps.

The vertical keel is partly continuous and partly intercostal, and extends from the stem nearly to the stern. The lower part is made of plates $11\frac{1}{2}$″ deep and $\frac{3}{8}$″ thick, worked intercostally between the frames just described; these are secured by a single row of rivets to a *continuous* plate 2 feet deep and $\frac{3}{8}$″ thick, on the upper edge of which two angle bars, $3'' \times 3'' \times \frac{3}{8}''$ are riveted. Sections of the keel are shown in Plate IV., and a side elevation of a short length of it is given in Fig. 75, where *i* denotes the

FIG. 75.



intercostal plate fitted between the two continuous transverse frames *f*, *f*, and *c* represents the continuous portion of the keel. The butts of this continuous plate are joined by double butt straps $\frac{1}{4}''$ thick, riveted as indicated at *b*; and the single angle bar, $3'' \times 3'' \times \frac{3}{8}''$, which secures the vertical and flat keel plates together, has its horizontal flange continuous, whilst the vertical one is cut away at the transverse frames, *f*, *f*, to allow these latter to cross the keel. Connection is made between the intercostal plates and continuous frames by short pieces of angle bar $2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}''$, marked *g* in the figure.

The first and second longitudinals from keel extend over about one-half the length of the ship, and are constructed somewhat similarly to the vertical keel, the main differences being that, in the longitudinals, lightening holes are cut, and *double* chain riveting is adopted for the butt straps of their continuous plates.

Returning now to a description of the transverse frames below and beyond the protective deck, those placed alternately between the ones already described are shown in Figs. 76 and 77. They are worked intercostally from the vertical keel to the 2nd longitudinal, and continuously above it. From the turn of the bilge upward they each consist of frame and reverse frame angle bars, the former being $8'' \times 3'' \times \frac{3}{8}''$ for those below the protective deck and $6'' \times 3'' \times \frac{3}{8}''$ elsewhere, and the latter $2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}''$. Below the bilges these frames are made of *floor plates* $\frac{9}{16}''$ thick, projecting sufficiently above the longitudinals to take hold of two continuous angle bars $2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}$, one of these being a prolongation of the reverse angle bar on the upper part of the frame, as indicated in Plate IV. The floor plates are connected to the vertical keel and longitudinals by angle bars $2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}''$ (*see* at *d*, Fig. 75), and to the bottom plating by $3'' \times 3'' \times \frac{3}{8}''$ angles; and the one above the 2nd longitudinal laps

Fig 76

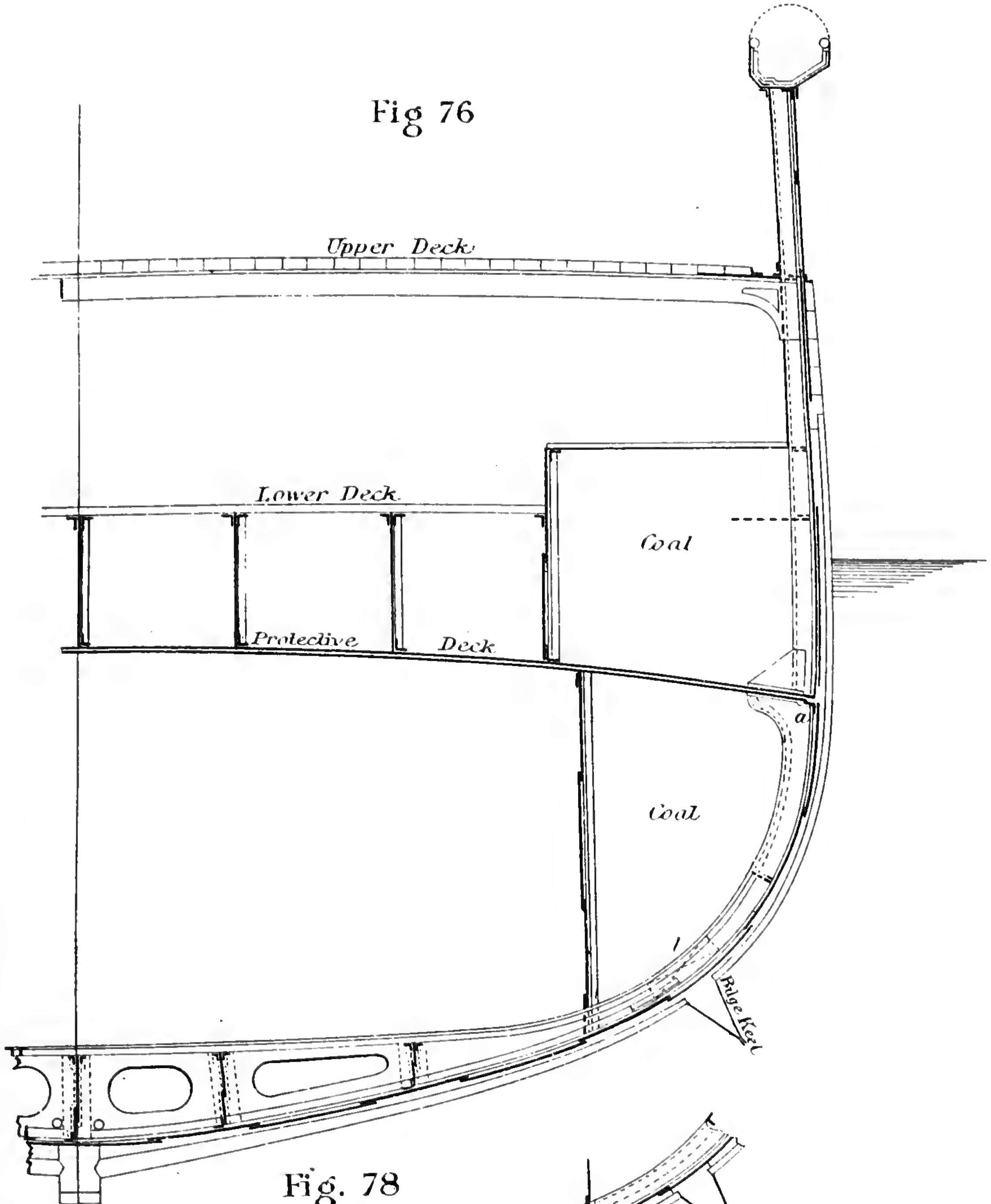


Fig. 77

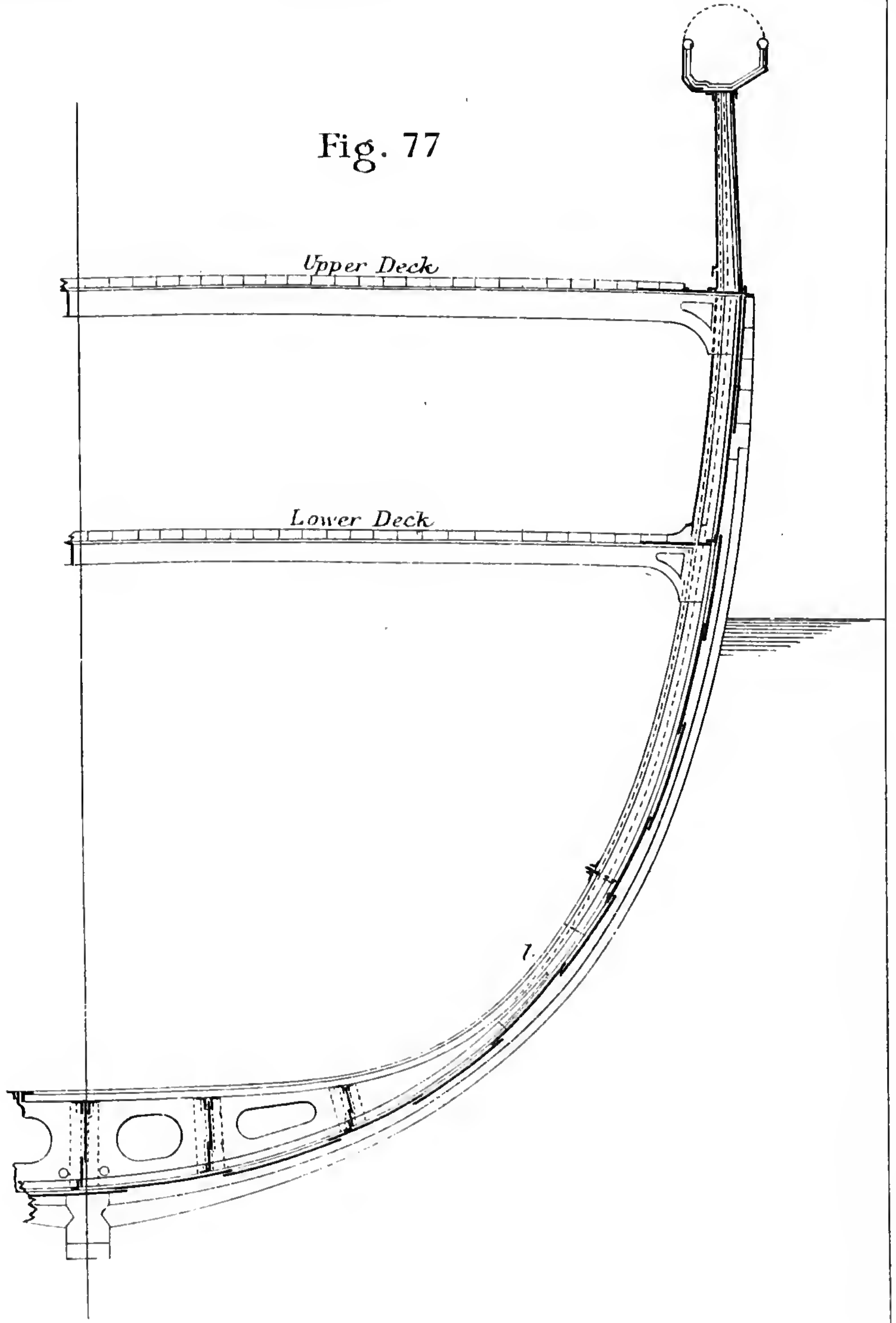
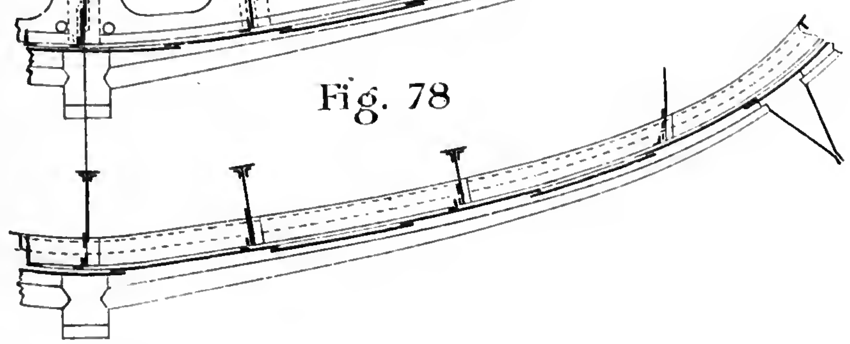


Fig. 78



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upon and is securely riveted to the frame angle bar, the junction being indicated at *l* in Figs. 76 and 77. The heads of these frames below the protective deck are also turned and secured to that deck as already mentioned.

The transverse frames *above* the protective deck are, in the later vessels of this class, made of Z-bars, $6'' \times 3\frac{1}{2}'' \times 3'' \times \frac{3}{8}''$, and secured to that deck by a bent plate, as shown in the midship section. The earlier vessels had these frames each made of a frame angle bar, $8'' \times 3'' \times \frac{3}{8}''$, and a reverse bar, $2\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{5}{16}''$, the heel being turned for security to the deck, similarly to the heads of the frames below.

Lastly, the 3rd and 4th longitudinals from keel on each side of the ship are made up of plates worked intercostally between all the transverse frames. The former are incorporated with the coal bunker bulkheads for the greater part of their length, and are, like the 4th longitudinals, secured to the outer bottom plating by short pieces of angle bar. As can be seen from the figures, the inner edges of the 4th longitudinals are fastened to the inner flanges of the transverse frames by two continuous angle bars.

The only part in the arrangement of beams and pillars calling for special remark is that in connection with giving support to the protective deck. Instead of working beams, longitudinal girders $3\frac{3}{4}$ feet deep and $\frac{3}{8}''$ thick are placed *above* the deck and extend throughout its length. They are worked between the transverse bulkheads dividing the engine and boiler rooms, their ends being secured thereto by short angle bars. Each girder is attached to the protective deck by a single angle bar, whilst two bars are riveted to its upper edge, to which the wood lower deck is afterwards secured (*see* Fig. 76). This omission of beams below the protective deck gives additional room for the engines and boilers.

The poop and forecastle beams are of angle bulb 5″ deep; those to the upper deck are tee-bulb 9″ deep, and the beams under the lower deck—except where the girders mentioned above are utilised—are tee-bulb 7″ deep.

The pillars vary in size from 5″ diameter and $\frac{3}{8}''$ thick to 3″ diameter and $\frac{3}{16}''$ thick, the smaller ones being used beneath the poop and forecastle.

3. *Of a Composite Sloop.*

On Plate V. are given two sections of the vessel depicted in Fig. 53; one taken amidships, and the other towards the ends.

The flat keel plate is 21 inches wide amidships and $\frac{3}{8}''$ thick, and is continuous from stem to stern.

The transverse frames are 20 inches apart throughout the ship, and are continuous from topside to topside across the keel, except those in wake of the watertight deck over engines and boilers, which are cut off by it.

The frames below the watertight deck consist of Z-bars $6'' \times 3\frac{1}{2}'' \times 3'' \times \frac{3}{8}''$, each having its lower end split into two angle bars and opened out to receive a floor plate $\frac{5}{16}''$ thick which runs continuously from the turn of the bilge on one side to the corresponding position on the other. The upper and lower angle bars of the frames on opposite sides of the middle line are joined by covering straps, the former being butted at the middle line as at *a* Fig. 80, and the latter on one side of it as shown at *b*, consecutive frames having this joint on opposite sides of the middle line. The lower angle bars are securely riveted to the flat keel plate, and the upper ends of these frames are connected to the watertight deck by bent bracket plates as indicated in the midship section.

Above and beyond the watertight deck the frames are of Z-bars $4'' \times 3'' \times 2\frac{1}{2}'' \times \frac{3}{8}''$, the former being secured to that deck, at their lower ends, by bent bracket plates. As shown in Fig. 81, each of the frames beyond the watertight deck has its inner flange cut off below the bilges, and is converted into an angle bar $3'' \times 3'' \times \frac{3}{8}''$, whilst on the upper edge of the floor plate, and extending from bilge to bilge, a reverse bar $2\frac{1}{4}'' \times 2\frac{1}{4}'' \times \frac{1}{4}''$ is riveted.

In the earlier composite vessels, frame and reverse frame angle bars were used where Z-bar frames are now employed.

FIG. 79.

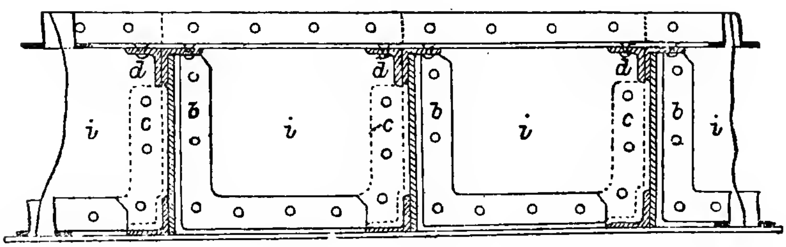


Plate V.

Fig. 80

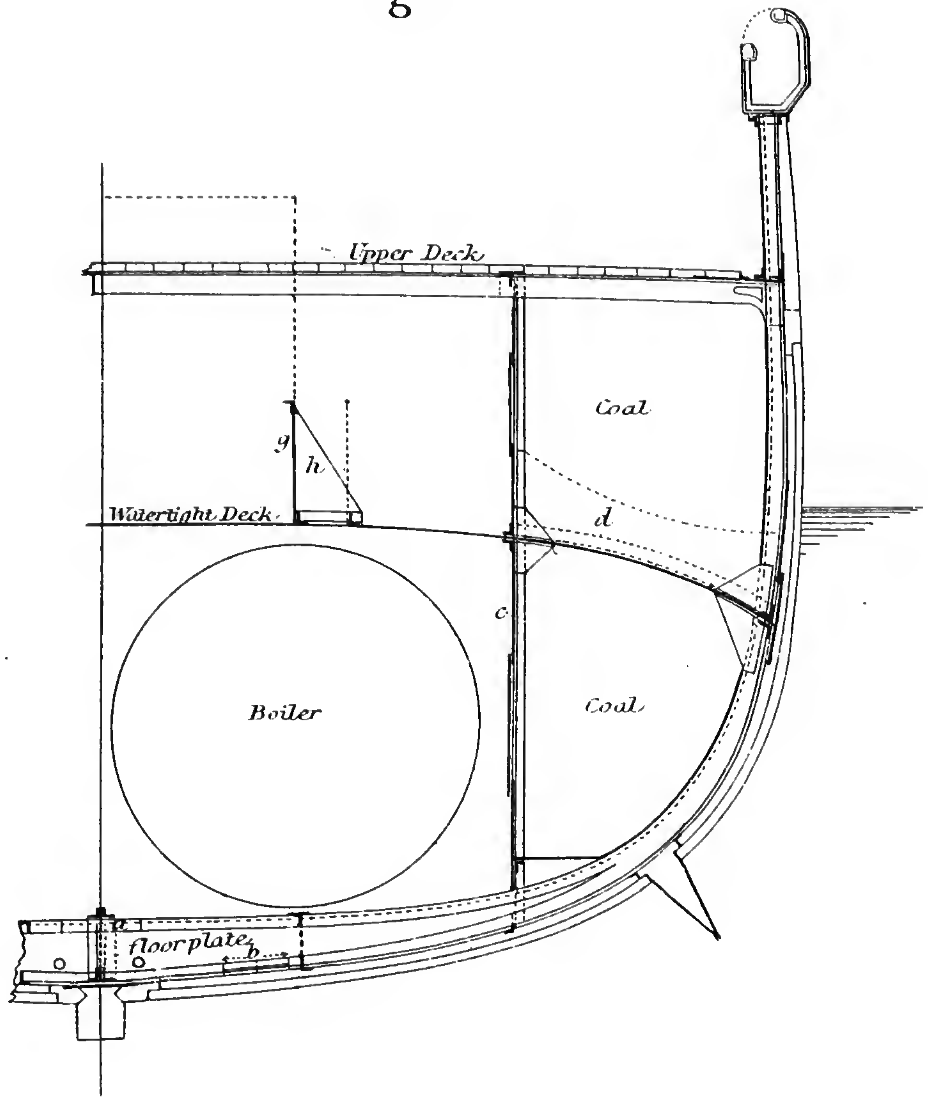
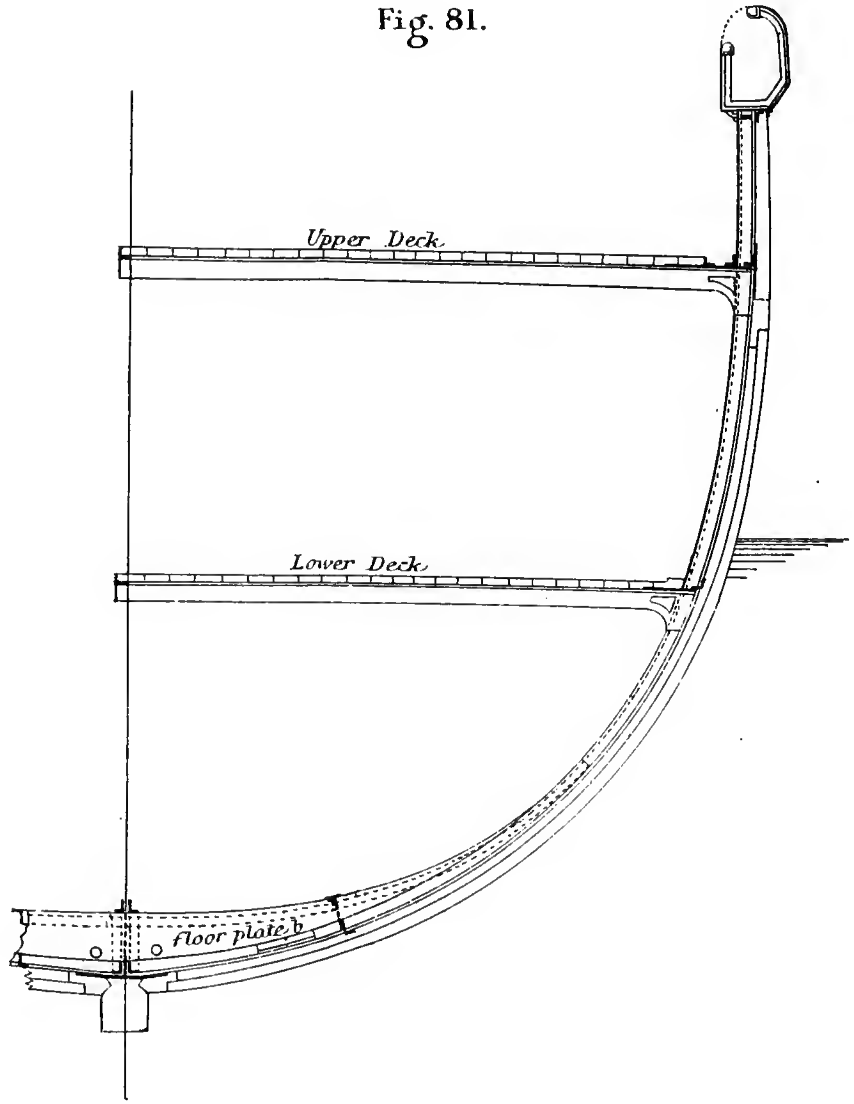


Fig. 81.



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The longitudinal framing resting upon the flat keel is known as the *intercostal keelson ;* it extends from stem to stern, and is made up of plates 18 inches deep and $\frac{3}{8}''$ thick fitted between the transverse frames, and projecting sufficiently far above them to be connected to two continuous angle bars $3'' \times 3'' \times \frac{3}{8}''$ which are secured to the inner angle bars of the frames. The flat keel and transverse frames are joined to the intercostal keelson by *half staple angle bars* $3'' \times 3'' \times \frac{3}{8}''$ placed on each side. This arrangement may be seen in section on Plate V., and is given in side elevation in Fig. 79, where the intercostal plates are marked *i*, the half staple angle bars on one side *b*, and those on the other side *c*. The short angle straps to the upper frame bars (page 96) are lettered *d*.

A longitudinal is worked on each side of the ship under the engines and boilers, and as far forward and aft as possible ; they are each formed of intercostal plates $\frac{5}{16}''$ thick secured to a continuous piece of plate—which is riveted to the inner bars of the frames—by short angle bars, and to the outer frame bars by an angle which has its outer flange continuous, and its inner one cut away at the transverse frames.

The steel watertight deck is supported on each side by a longitudinal girder *g* (Fig. 80), which is stiffened at intervals by brackets as shown at *h ;* also by the coal bunker bulkhead *c*, and, between that bulkhead and the ship's side, by several transverse brackets composed of plates and angle bars, one of these being shown at *d*. The beams under the poop and forecastle, as well as those to the lower deck, are of angle bulb bars 5″ deep ; the upper deck angle bulb beams are $6\frac{1}{2}''$ deep.

The pillars in these vessels are small, varying from 2 to 3 inches in diameter, and are $\frac{1}{8}''$ thick.

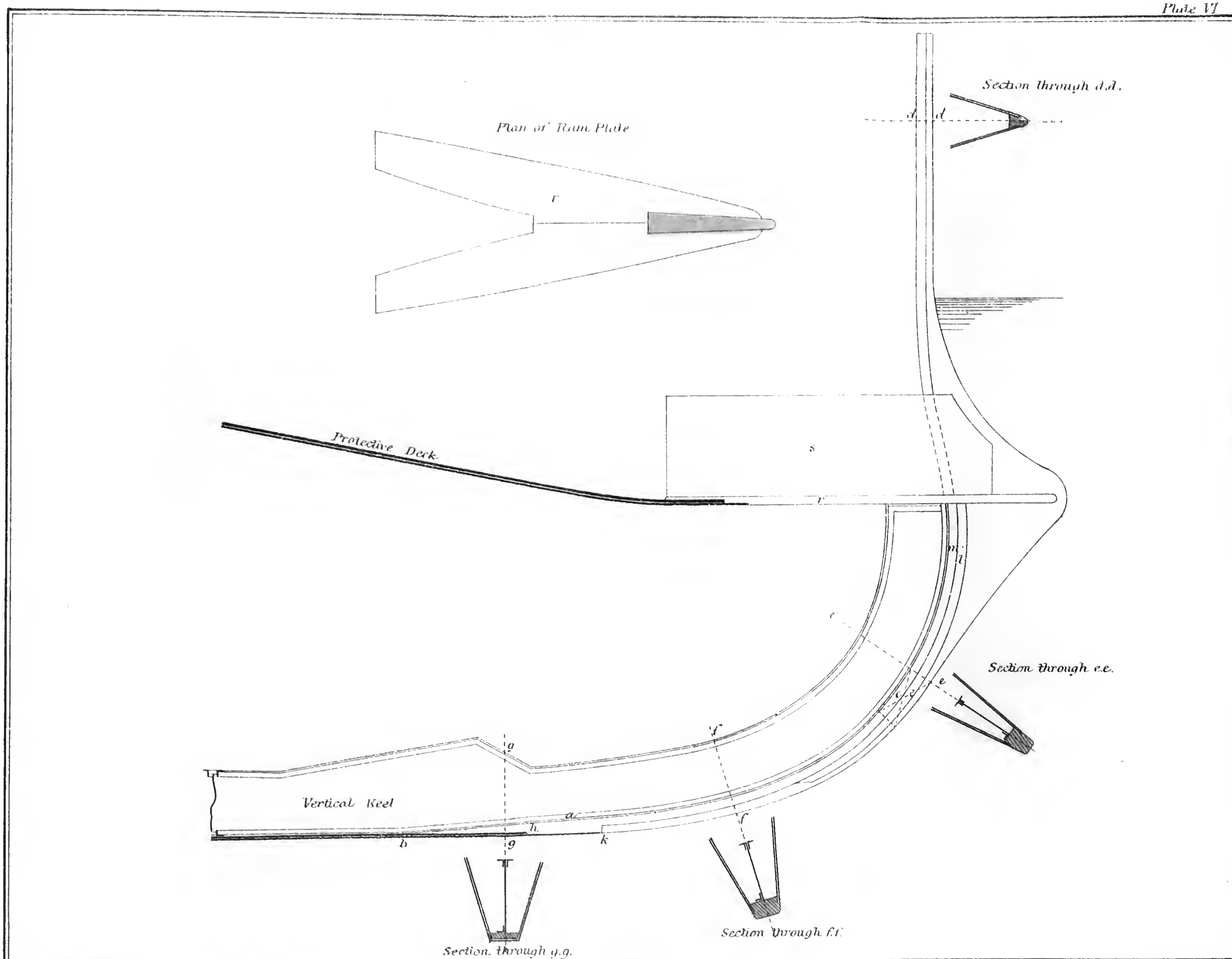
CHAPTER VII.

STEMS AND STERNPOSTS.

THE stems of all ships complete, as it were, the framing at the fore part of the vessel, and the sternpost of a twin-screw ship helps to perform a similar function at the after part. Where a single screw only is employed, there are two posts aft; the fore or *body post* forming a part of the stern boundary of the framing, and the after or *rudder post*, to which the rudder is hung.

Iron forgings, or, more recently, steel castings, are most generally used for these parts of a ship, but their employment in sheathed or composite vessels is inadmissible on account of the galvanic action which would be almost certainly set up betweem them and the copper sheathing, and which would result in the rapid corrosion of the iron or steel, since these latter are *electro-positive* to copper. For this reason composite ships have their stems and sternposts made of wood ; but as this material would not be sufficiently strong for the larger sheathed ships, the stems and body posts of these vessels are made of some metallic alloy which is electro-negative to the copper, so that any galvanic action which ensues will only assist the dissolution of the latter and thus increase its antifouling properties. *Gunmetal* (composed of copper and tin) is often utilised for this purpose, or some other material of similar composition, such as phosphor bronze ; and as their strength and ductility are inferior to that of iron or steel, the castings must be made larger and heavier than would be necessary were the latter materials used. It may be added that in both sheathed and composite vessels other under-water fittings are also made of gun-metal or phosphor bronze, including the struts or shaft brackets on the outside of twin-screw ships which support the ends of the propeller shafts.

Plate VI. represents the stem of the armour-clad shown in Fig. 51, and the manner in which it is connected to the keel, &c. It extends from the height of the upper deck to the point *b*, and is made in two parts, which are joined together at *c* by a number of through rivets about 2″ in diameter. Below water the stem is formed into a spur or *ram*, with a view to running into, and piercing the thin bottom plating of, an enemy's ship.


Section through d.d.
d
d
Plan of Ram Plate
r
s
Protective Deck
r
m
l
e
e
c
c
Section through e.e.
f
g
Vertical Keel
a
f
h
b
g
k
Section through g.g.
Section through f.f.

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In order that this operation may be carried out without serious damage to the bow of the attacking vessel, the stem must be well supported against being driven bodily backwards, and also strengthened to resist the *wrenching* stresses which would be induced on striking the side of a vessel under weigh, or by ramming obliquely. The latter stresses are principally provided against by working at the bow a *horizontal ram plate r*, 5″ thick, shaped to receive the stem. It is in some ships made in one piece, whilst in others it consists of two parts, which are connected together in the middle line of the ship, as shown in plan on Plate VI. This ram plate projects outside the general surface of the bottom, and is securely fastened thereto and to the stem by stout angle bars both inside and outside the ship. The angular spaces between the projecting parts of the ram plate and the outside of the vessel are filled up with wood, to prevent the anchors becoming fouled there; and this wood is afterwards covered with a thin steel plate to shield it from damage. The stem is strengthened against being driven backwards by the protective deck, which runs down to meet the ram plate; by the bottom and side plating, which is terminated at recesses or *rabbets* cut into the stem, as shown at *l* and *m* respectively in elevation and in the sections given at various parts of the stem; by horizontal framing known as *breast-hooks* (not indicated in the figure), each consisting of a thin plate joined to both sides of the ship and to the stem by an angle bar secured round its edge; and, lastly, by a 2″ armour plate *s* riveted to the stem and bottom plating above the ram plate outside the ship on each side. It need scarcely be remarked that these direct supports are also of considerable value for resisting wrenching stresses.

The stem is secured to the keel plates in the following manner. The vertical keel runs on the inside of the stem up to the horizontal ram plate, as does also one of the lower angle bars (*a*), and this latter is tap-riveted to the stem. The vertical keel is also attached at its upper end to the ram plate. The lower or horizontal parts of the flat keel plates terminate at stops formed in the stem at *h* and *k* respectively; the other parts of these plates are continued along the side of the stem into the rabbets *l* and *m*, and are then incorporated with the ordinary bottom plating. Both flat keel plates are well connected to the stem by tap rivets.

It may be mentioned here that many of the earlier battle ships of the Royal Navy have a belt of side armour which extends from end to end, and their stems must therefore be rabbeted to receive the ends of the armour plates; this would provide efficient support to the stem in the event of the ship being used as a ram.

The sternpost of the armour-clad is much lighter and simpler in construction than the stem, and is shown in side elevation in Plate VII. It is made in one length from *a* to *b*, the lower horizontal part virtually forming the after end of the keel. The contour of the stern is completed above *a* by bent plates, shaped as shown in the sections. The vertical keel runs as far as *d* on the sternpost, and one of its lower angle bars are tap-riveted thereto. The inner flat keel ends at *e*, the part of the post before that point being so planed that a stop is formed against which the plate abuts. The bottom of the outer plate finishes at *f*, and the sides run along to the upright part of the post, against which it laps, after stepping up over the stop *s* (*see* section at that part) forged on each side for convenience in ending the plating. Sometimes the sides of the *inner* plate are continued to the end, no stop such as *e* being then necessary; and the outer plate in this case terminates at *f*.

No rabbets are taken out of the sternpost for the reception of the bottom plating, the latter being simply lapped upon the former and secured by tap rivets. The rudder head passes through the upper part of the post, and the pintles of the rudder are received into the lugs or gudgeons *g* which are formed on the sternpost (*see* Chap. XIII.).

Figs. 82 and 83 of Plate VIII. are representations of the stem and sternposts, respectively, of the sheathed corvette outlined in Fig. 52. The stem extends from the height of the forecastle deck to the point *a*, where it is joined to a second casting *b*, the after end of which is formed to take the fore end of the wood keel, whilst the vertical and flat keel plates are connected to *b* and to the stem. The sections also clearly indicate how the ends of the plating and planking are housed in the rabbets *e*, *f*, and *g*.

It will be noticed that the body post is a rather complicated casting, this being brought about partly through the necessity for receiving the propeller shaft, and partly because rabbets have to be formed in it to receive the ends of the bottom planking. The upper end *h* of this body post is firmly riveted to a steel flat inside the ship, and its lower end is united to a metal casting the forward and after ends of which take hold of the end of the wood keel and lower part of rudder post respectively. The mode of securing the heel of the body post is shown in side and end elevations at

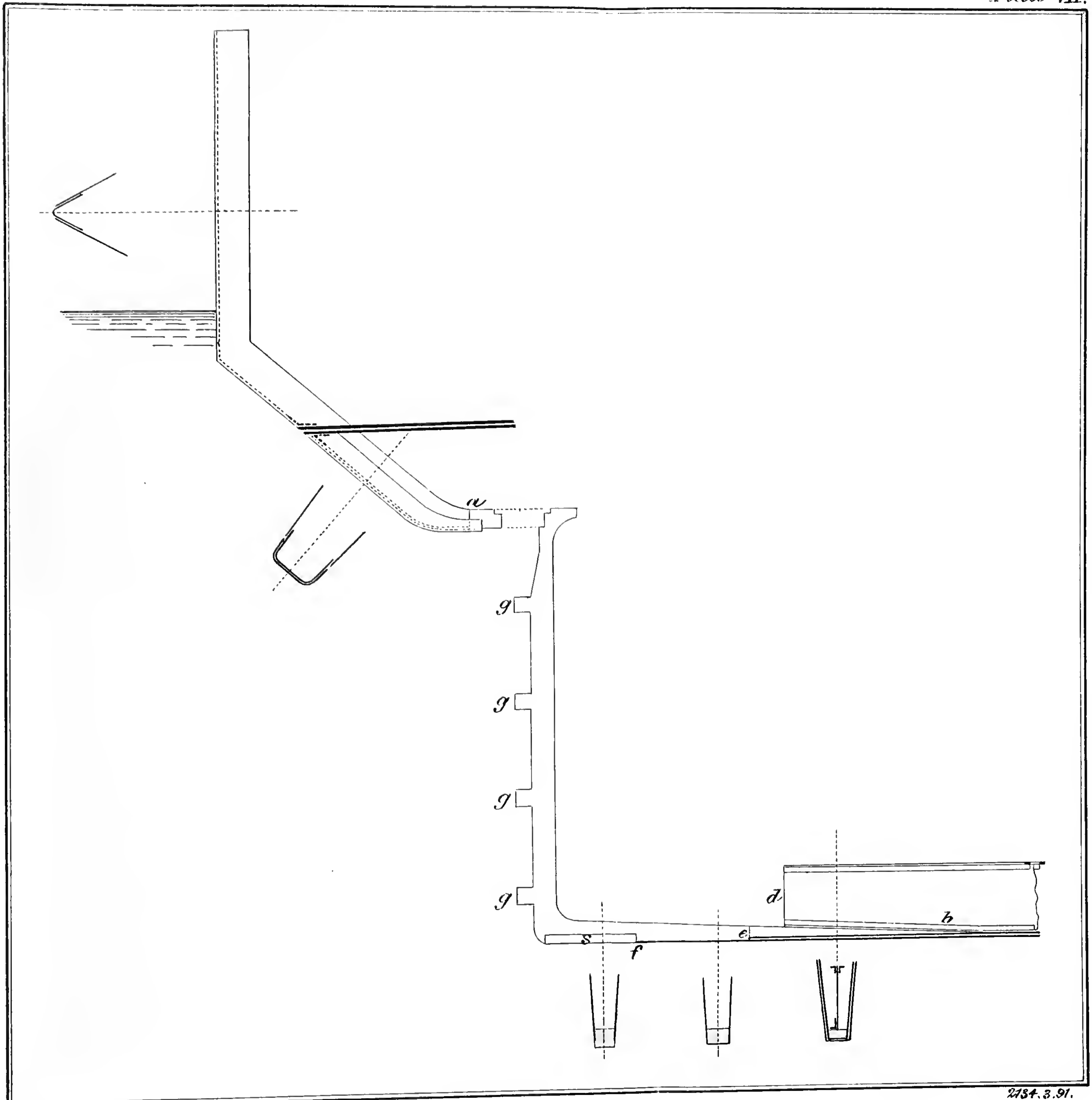

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Fig. 83.

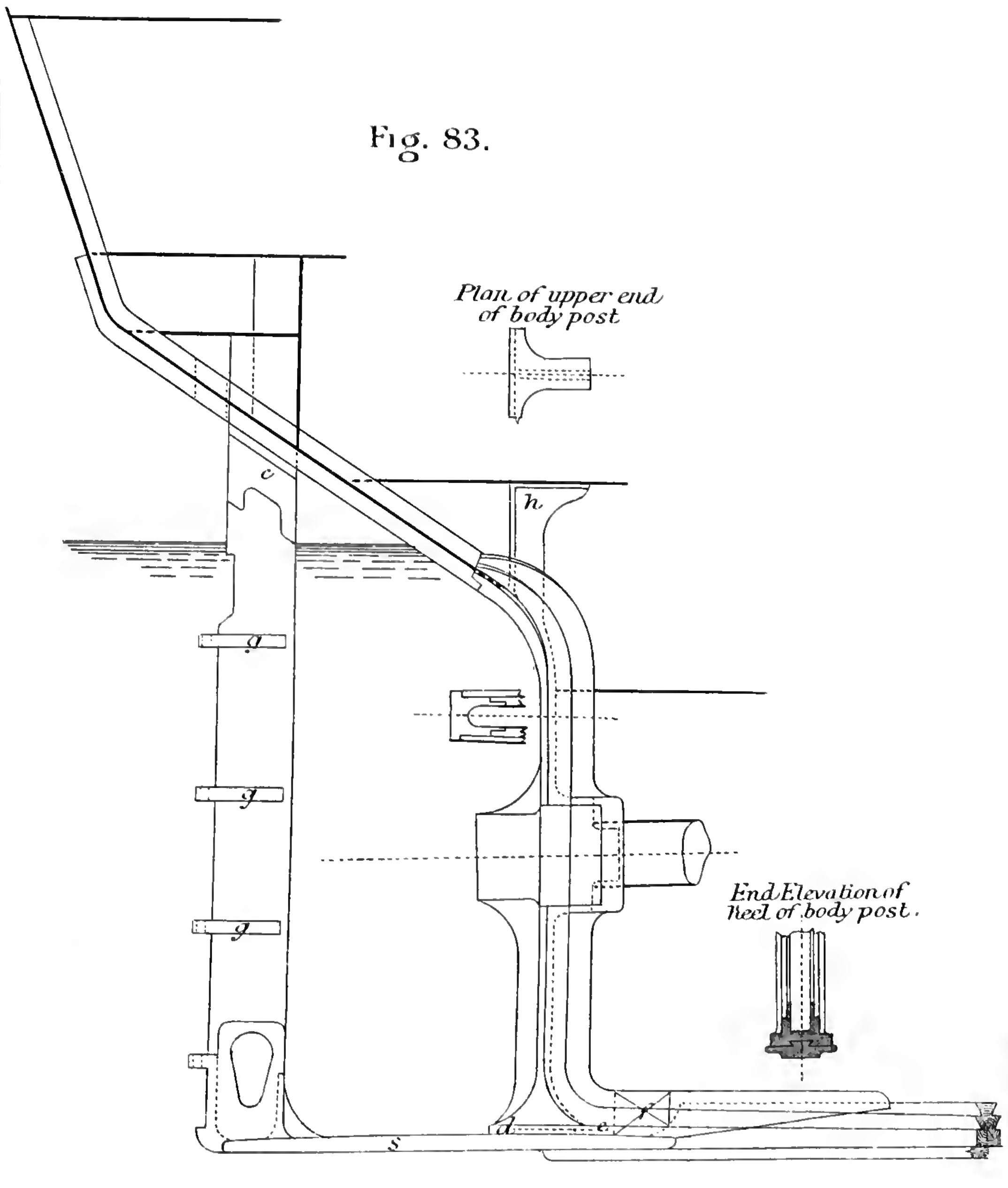

Fig. 82

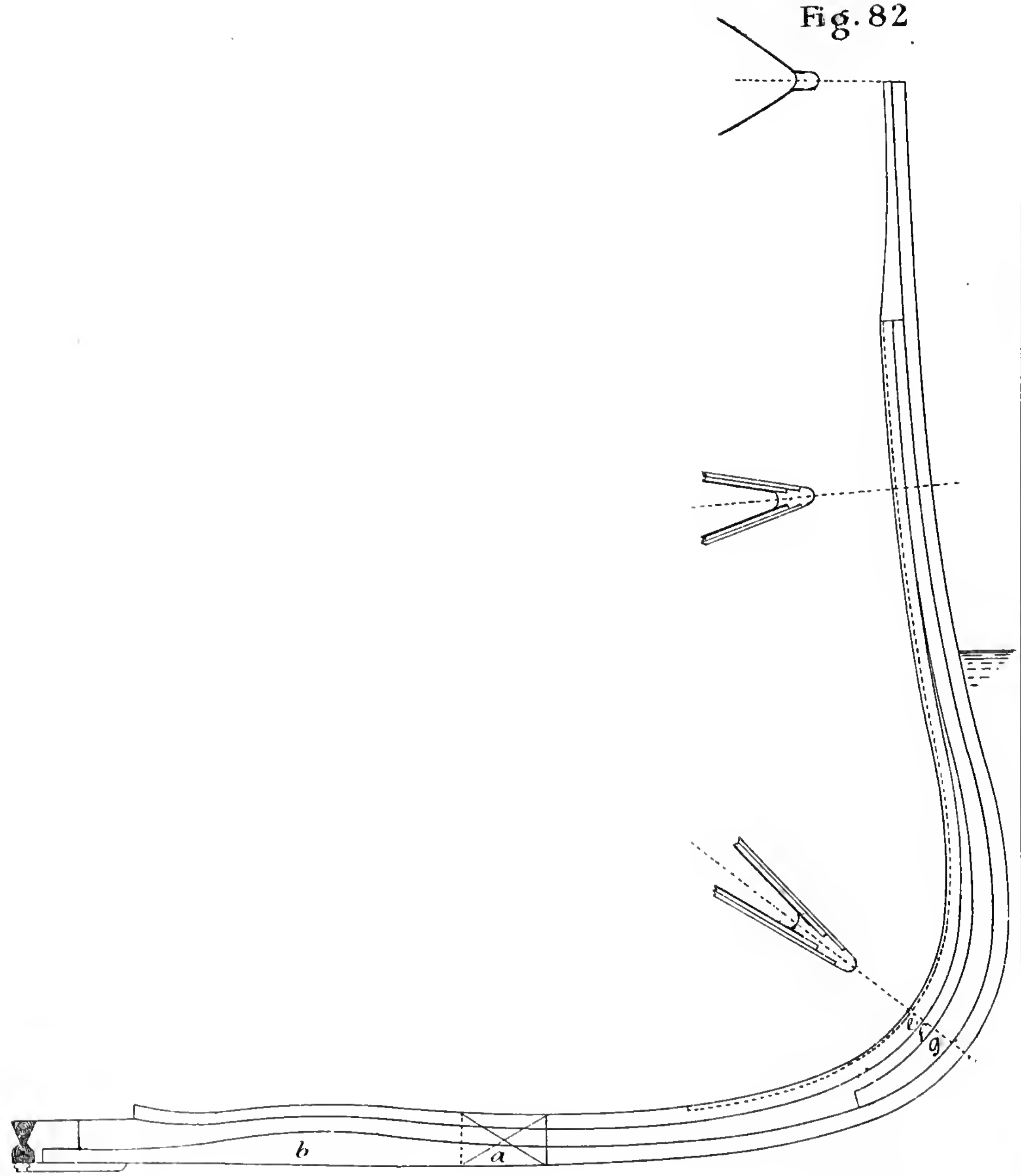

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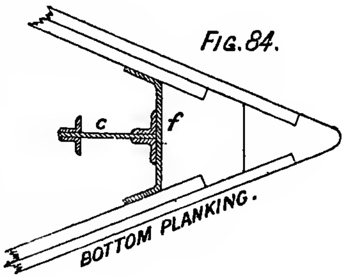
Fig. 83. On the lower casting a projection (*d e* in side, and *k* in end, elevation) is formed, which fits into a corresponding groove in the post, thus dovetailing these parts together; and in addition to this connection, the lower casting and post are lapped and through riveted together on each side of the ship at the parts marked *f*, besides being well tap riveted together at the bottom of the body post. The middle part *s* of the lower casting is called the *sole piece*, and is made broad and flat; the former in order to ensure lateral rigidity, and the latter to provide as much room as possible for the propeller. The rudder post is made of wood, its upper end being received between and securely bolted to plates worked vertically inside the ship; it is also attached to the bottom planking by a metal casting *c* on each side. The metal braces *g* are firmly bolted to the post, and carry the rudder.

A horizontal section through the wood stem of a composite vessel is given in Fig. 84. The various pieces of which it is made are bolted to a flanged plate *f*, which is virtually a continuation of the flat keel, and this in turn is riveted to double angle bars on the outer edge of a plate *c* which extends continuously up the stem from the intercostal keelson, and has double angle bars also upon its inner edge. The wood sternpost is secured at its heel to the wood keel by bolts driven through a metal casting fitted at their junction, and is closely incorporated with the *deadwood* aft, *i.e.*, the mass of timber necessarily worked at the narrow underwater part of the stern.

FIG. 84.



CHAPTER VIII.

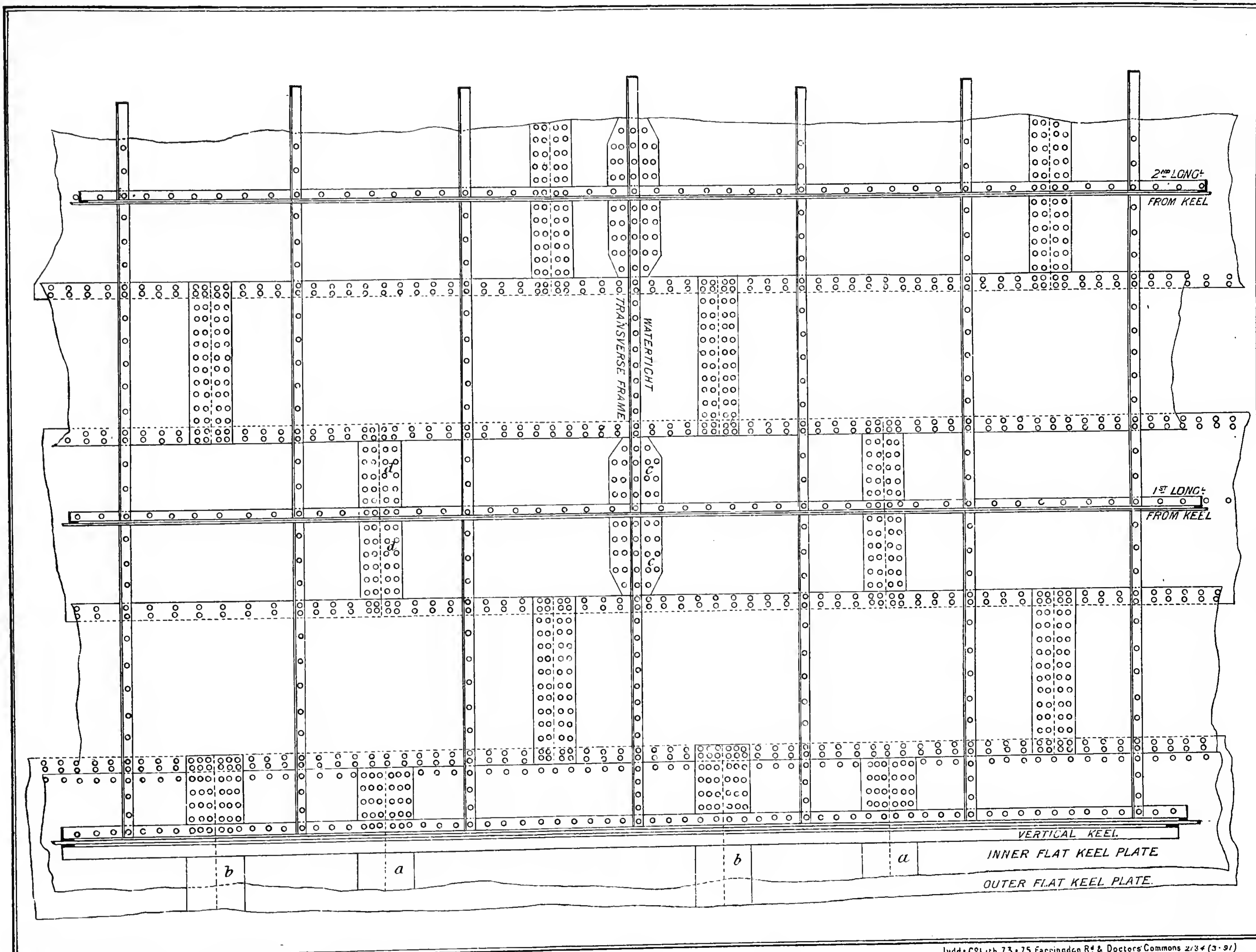
BOTTOM AND SIDE PLATING AND PLANKING.

The bottom and side plating of all ships is arranged in longitudinal layers or *strakes*, consecutive plates in each strake being joined together by butt straps. Adjacent strakes of the *outer* bottom plating are always connected to each other along their edges by lap joints, as indicated in the sections on Plates III. and IV., alternate strakes fitting closely against the frames whilst the others are separated therefrom by the thickness of the plates. This arrangement is known as the "raised and sunken strake system" of plating. In recent ships provided with a double bottom, the *inner* skin has also been worked as just described, but in earlier vessels the edges were made flush, as in Fig. 73, and were joined by edge strips (*see* Chapter IV.). Again, the side plating above the water line is, in some ships, worked with flush edges, for the sake of appearance, whereas in others the ordinary lapped system obtains for this plating also, as will be noticed in Fig. 74. It will be seen from the same figure, and from Fig. 76, that the plating below the protective decks is cut off from that above them. This is because it was thought probable that a shot leaving the ship above the deck, at the side opposite to that which it entered, would tear a large area of plating from the frames, and thus, if this were not disconnected from the plating below, would open up compartments beneath the protective deck to the sea. In the latest vessels, however, this has not been done, it being considered that the extra work involved more than counterbalances any possible gain that may accrue from working the plating as just described. The skin plating on the frames behind armour of armour-clads is always worked with flush edges and butts.

Confining attention for the present to the armour-clad, the lowest strakes of outer bottom plating are formed by the flat keel plates, the sizes of which have been already mentioned in Chapter VI. The next strake from keel on each side is a "sunken" one, upon which the outer keel plate laps; and from thence raised and sunken strakes occur alternately, the former fitting against the longitudinal frames (*see* Fig. 73). The plates employed are from 12 to 16 feet long, about 4 feet wide, and $\frac{9}{16}''$ thick. Their butts are placed midway between the transverse frames as shown in Plate IX., which gives a plan of part of the outer bottom plating, &c., and indicates how the butts in the several strakes are arranged with reference to each other, or *shifted* as it is termed. As the butts are places of relative weakness, it is obviously advantageous to have as few of these as possible in the same transverse section, and thus the proper shift of the butts is a question of importance. The disposition adopted in any case depends upon the length of the plates used in relation to the frame spacing. That shown in Plate IX. is a very usual arrangement of butts—known as the *diagonal* shift—and is employed when the length of each plate is three times the distance between transverse frames. In this arrangement of bottom plating a butt strap is fitted between any given pair of transverse frames at every third strake; and with longer plates other shifts may be adopted to give a still smaller proportion of butted strakes in the same frame space. The butt straps are single and worked inside the ship. Those marked *a* and *b* join the butts of the inner and outer flat keel plates respectively, and they extend from the vertical keel angle bars on each side to the edges of the plates they connect. The straps of the sunken strakes of bottom plating run the whole breadth of the plate, whereas those to the raised ones reach from the angle bars of the longitudinals to the edges of the adjacent sunken strakes, as at *d*, *d*. Treble chain riveting is employed in the butts of flat keel, whilst the other butts of plating are double chain riveted. The edges of the inner flat keel plate are each secured to the outer plate by a single row of rivets, but all the other edges of the outside plating are double chain riveted, with the exception of those above the protective deck and armour belt, which are only single riveted. The rivets are $\frac{7}{8}''$ in diameter—except those in the keel, which are rather larger—and have pan heads and countersunk points, the heads of course being inside the ship. As the butts and laps must be caulked to make the plating watertight, these rivets are closely spaced, and the ends of the several plates are accurately fitted together in order that the caulking may be efficient at those parts.

Single rows of rivets connect the bottom plating to the transverse and longitudinal frames, as shown in Plate IX., the spaces between the raised strakes and the transverse frames being each filled with a *liner*, or piece of plate equal in thickness to the bottom plating, and usually of the same width as the angle bar of the frame. Where, however, connection is made to transverse watertight frames, the rivets must be put close together to admit of the frames being caulked. At those sections, therefore, the plating is pierced with a larger number of holes than usual, and so is correspondingly weakened. To compensate for this, the liners to the raised strakes of plating in wake of a watertight frame are made sufficiently wide to take two rows of rivets on each side, thus acting as a butt strap across the weakened section. One of these liners is marked *c* in the Plate.

The outside plating is made thicker in certain special parts for protective and other purposes. For example, that in front of the guns between the upper and spar decks is in two thicknesses of $\frac{1}{2}''$ each, so arranged that the plates in one thickness form edge strips and butt straps to those in the other. These plates are screwed together, the connecting screws being made flush with the inner and outer surfaces of the plating. Ordinary rivets are not used here because of the liability of their heads to fly across the gun deck (where a number of men must necessarily be employed in action) if the side should be struck. The plating at the bow is also doubled for a length of 40 or 50 feet, plates being worked beneath some of the raised strakes as well as outside the sunken strakes, so forming a flush outer surface. This additional plating provides against the wear and tear due to the rubbing of the anchors and cables at this part, and also helps to support the stem, besides diminishing the probability of the bow plating being broken through, after ramming an enemy's ship, by the jagged edges of the hole made in the vessel attacked. Each thickness of plating is double tap riveted to the stem after it has been fitted into the rabbet prepared to receive it, and similar fastenings are used in the single thickness of plating which laps upon the sternpost. Again, the skin plating behind armour is in two $\frac{1}{2}''$ thicknesses, each thickness being arranged to act as butt straps and edge strips to the other. On the stronger sides of recent first-class battle ships (*see* Fig. 63) the combined thickness of this plating is $2\frac{1}{4}$ inches.



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The inner bottom extends transversely to the first longitudinal below the shelf plate, and to it the heels of the main frames behind armour are each secured by a short narrow piece of plate as indicated in the midship section. This bottom plating is rather less than $\frac{3}{8}''$ in mean thickness, is single riveted at the edges, and double riveted at the butts; this plating is also riveted to each frame in a similar manner to that of the outer bottom.

It will be noticed that the *wing passage bulkhead* (Fig 73), placed several feet from the side of the ship, virtually forms a continuation of the inner bottom up to the main deck, and therefore there are, with the *coal bunker bulkhead*, two inner bottoms at the side, forming a great protection if the outer skin should be broken through by a ram or torpedo.

To avoid sudden discontinuity of strength at the ends of the inner bottom, due to the omission beyond those ends of the inner bottom plating and the employment of weaker longitudinals, the continuous longitudinal frames in the double bottom space are prolonged beyond the ends of that space for a short distance, and are each tapered down to the width of, and firmly secured to, the corresponding Z-bar longitudinal; the inner bottom plating worked on the longitudinal is also extended for the same distance. The strake of inside plating on the vertical keel is continued towards the ends of the ship as a *gutter plate* (*see* Fig. 74), which connects the transverse frames on opposite sides of the keel.

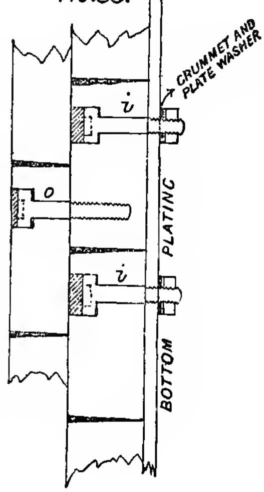
Passing now to the sheathed ships, the butts of the flat keel plate are double chain riveted, whilst the plating of bottom is worked in a manner similar to that described for the outside plating of armour-clads, except that the edges are all single riveted only, in view of the support given by the planking. Most of the plating is about $\frac{3}{8}''$ thick, but in the vicinity of the water line it is $\frac{7}{8}''$ for additional protection against gun attack, worked in two thicknesses; and the plating against the upper deck beam arms (the *sheer strakes* to upper deck) is also doubled from considerations of strength (*see* Chap. XVI.)

The wood keel is 14″ wide, and is securely fastened to the flat keel plate by $\frac{7}{8}''$ copper bolts. It is made of *elm*, a very tough wood and much twisted in grain, and thus is not liable to split when the bolts are driven through it. Elm is also very durable when wholly immersed in water. A *false keel*, also of elm, is lightly attached by metal nails to the main keel, as it protects the latter from injury should the vessel take the ground, any damaged part of the *false* keel being easily replaced. In sailing ships, this keel also diminishes the tendency to move sideways away from the wind.

In arranging and fastening the bottom planking every precaution has been taken to *insulate* the copper sheathing from the iron or steel of the hull, because if they are *in metallic connection in the same water*, the galvanic action resulting will injuriously affect the hull. For this reason the planking is worked in two thicknesses or layers—except on the topsides where one thickness only is used—the inner one being secured to the bottom plating whilst the outer thickness is fastened to the inner.

The planks run longitudinally, and are so disposed that each one of either thickness acts as an edge strip to planks in the other, so securing an edge connection. The inner layer is of *teak* this wood containing a resinous oil which prevents the rusting of iron when in contact with the latter. Teak is also often employed for the outer thickness, although others are sometimes used below the water line; and the thicker strakes (*or garboards*) near the keel are of elm. The inner layer is 3″ thick and is secured to the steel skin by $\frac{3}{4}''$ iron screw bolts, carefully screwed through the plating and provided with a nut inside the ship as an additional security, as shown at *i* in the part section of planking given at Fig. 85; three of these bolts are put in each plank between consecutive transverse frames, disposed as indicated in front elevation in Fig. 86. To make each bolt hole watertight, a ring of hemp called a *grummet* is steeped in red lead and put underneath a thin plate washer, against which the nut is tightly hove up.

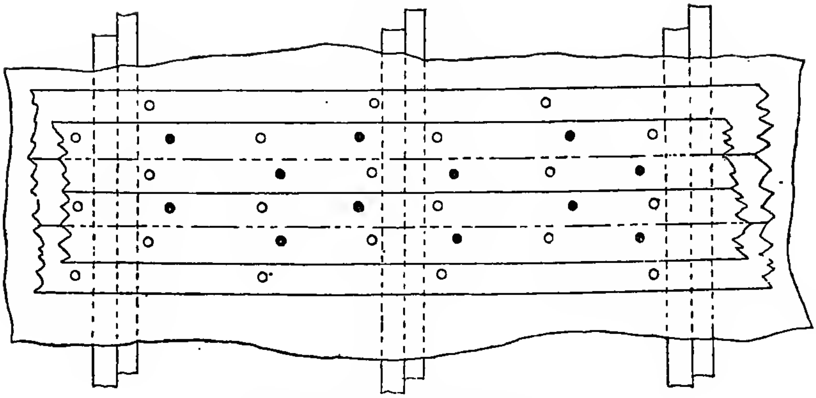
FIG. 85.



A hempen grummet is also placed beneath the head of each bolt, and the holes above the heads are filled up with a mixture of red and white lead.

The outer planking is $2\frac{1}{2}''$ thick and is screwed to the inner thickness by $\frac{5}{8}''$ *naval brass* (62 parts copper, 37 zinc and 1 tin) screw bolts, much care being exercised to avoid making the holes for these bolts pass right through to the skin plating, and to see that the points of the bolts, when properly in place, are at least

FIG. 86.



$\frac{1}{2}''$ away from that plating; this is to prevent the possibility of metallic connection, through these bolts, between the steel hull and copper sheathing. One of these bolts is marked $\dot{o}$ in Fig. 85, and their disposition is given in Fig. 86.

The outer surface of the bottom plating is well painted with a mixture of red and white lead, and the two thicknesses of planking, where they are in contact with each other, are also well coated with the same materials. Both thicknesses of planking are caulked with oakum at the edges and butts of the planks, and the sheets of copper are nailed to the outer thickness, after tarred brown paper has been interposed between.

The above is the most efficient known method of preventing fouling, but the adoption of copper necessitates all the precautions for ensuring insulation just detailed, and also the stopping of the copper short of the stem, sternpost, &c., for the same reason; it involves the use of heavy and expensive metal stems, &c., as explained in Chap. VII., and there is likewise the danger in these ships that a few of the bottom planks may be stripped off by grounding or other means, thus exposing a small area of the bottom plating to the action of the large area of the copper, and leading to the rapid corrosion of the former; any such damage should therefore be repaired as soon as possible.

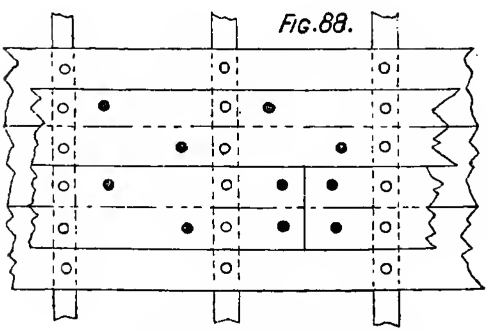
If a sheathing *electro-positive* to steel could be used, the before-mentioned precautions would be unnecessary, as any galvanic action which occurred would assist the dissolution of the sheathing and thus keep the surface freer from incrustation. This consideration led to the trial of *zinc*, when only *one* thickness of plank was used, which was not caulked, so that any water getting beneath the zinc had free access to the skin plating. Iron stems and sternposts were also employed, and means were taken to secure metallic connection between the zinc and hull. Great care was, however, needed to ensure that the zinc should wear evenly over its surface, and that the rate of wasting away should be as small as possible without injuring its antifouling qualities; and since these qualities are decidedly inferior to those of copper, the latter material is the only one now used.

Before leaving this part of the subject it may be mentioned that tests have shown the existence of metallic contact between the copper and hulls of sheathed ships, even after carefully working and fastening the planking as already described. The galvanic action resulting has not injuriously affected the hull, and, therefore, only one thickness of planking, carefully caulked, and fastened with metal bolts, has been employed in some recent ships of this class.

The plating on the outside of a composite ship is principally in the form of *sheer strakes* to the forecastle, poop and upper decks, some thicker plating being also worked in the vicinity of the water line (*see* Plate V.).

The planking is of teak, worked in two thicknesses for the same reasons as in the sheathed ship. The inner layer is 3″ and the outer one $2\frac{1}{2}''$ thick. Each plank of the inner thickness is fastened to each transverse frame by two $\frac{3}{4}''$ naval brass screw bolts screwed into the outer flange, each bolt being also provided

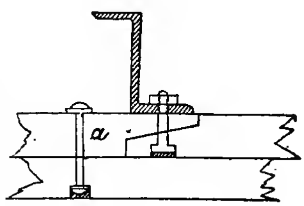
FIG. 88.



BOLTS OF INNER THICKNESS ....O

" " OUTER " ....●

FIG. 87.



with a nut on its point. The ends of these planks are supported on the frames, adjacent ones being formed into a *scarph*, as shown in plan in Fig. 87, so that they may be secured by the same bolts. The outer thickness is attached to the inner by $\frac{1}{2}''$ copper bolts, each having its point clenched upon a metal ring, and having a similar ring beneath its head ; one of these bolts is marked *a* in Fig. 87. The part elevation of the planking shown in Fig. 88 gives the distribution of the fastenings in both thicknesses.

The edges and butts of the planks are carefully caulked, and the copper is attached as before described for the sheathed vessel.

CHAPTER IX.

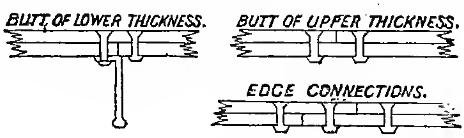
DECK PLATING AND PLANKING.

Deck plating is employed to protect vital parts of the ship from gun fire, to contribute to the strength of the vessel (*see* Chap. XVI.), to give additional local support under heavy weights, such as coals, guns, &c., or to form platforms with a view to dividing the vessel into watertight compartments.

The extent of the plating on the various decks of the ships under consideration may be seen from an inspection of the sections on Plates III. to V. The plating is arranged in longitudinal layers, except the strake nearest each side of the ship, which is laid parallel to that side and is known as a *stringer*. On some decks, as will be noticed, these constitute the whole of the plating worked; they help to connect the beams and side, and serve to stiffen the plating in their vicinity, besides assisting to augment the general strength of the ship.

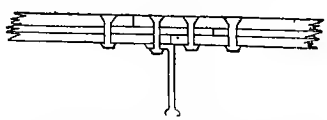
Protective decks are made up of two or three thicknesses of plating. In the former case each thickness is arranged to act as edge strips and butt straps to the other, single riveting being adopted. Fig. 89 shows how this is accomplished, the butts of the lower thickness resting upon beams, and those of the upper thickness midway between them, whilst the edges are so arranged that three rows of rivets suffice to secure them in both thicknesses.

FIG. 89.



When three thicknesses are used, the edges and butts of each are often so disposed with reference to the others that four rows of rivets are sufficient, as in Fig. 90.

FIG. 90.



This deck plating is also securely riveted to each beam, and its edges and butts are caulked.

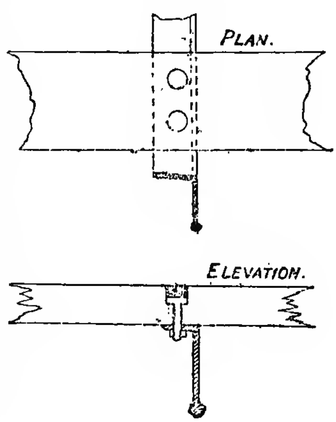
The plating on the remaining decks of all the vessels is in one thickness, varying from $\frac{1}{4}''$ to $\frac{1}{2}''$; and where several strakes are worked their edges are usually lapped and the ends of the plates butted, single riveting being employed for the former and double riveting in the latter. These plates are riveted to the beams and, when a complete iron or steel deck is worked, they have their edges and butts caulked for watertightness. Special care is taken to attain this latter result at the sides of the ship, by stopping the deck plating proper at the inside edge of the frames and fitting short pieces of plate between the frames; these pieces of plate lap upon and are riveted to the deck plating, and are united to the frames and bottom plating by angle bars having turned ends to fit closely against the frames, which angle bars are afterwards caulked. Where it is not necessary to make a stringer watertight, scores are cut into the plate so that it may be fitted over the frames, and connection is then made with the bottom or side plating by short angle bars.

Planking is laid over each tier of beams, even when these latter have been previously covered by plating. The reasons for this latter proceeding are that it conduces to the comfort of the crew, and prevents the surface of the plating from experiencing all the variations of temperature which occur in the external atmosphere, thus hindering the condensation of moisture on the under side of the deck, which would injure cargo or produce discomfort by dripping. It also stiffens the plating above which it is fitted, and so adds to the strength of the ship.

The wood employed is generally *Dantzic fir*, as it is tough, elastic, moderately hard, aud of light weight. In places subject to much wear and tear, as for example the parts rubbed by the chain cables, the very durable materials teak or oak are substituted for the fir; and on some decks teak alone is used.

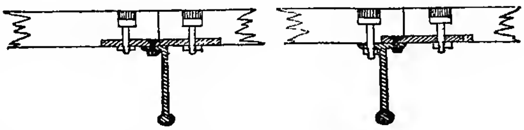
The planks are laid longitudinally (except in such places as on the longitudinal girders of the sheathed corvettes where they are necessarily laid transversely for support), and vary from a maximum thickness of 4″ to about $1\frac{1}{2}$ inches on the smallest platforms. When no plating is worked, each plank is secured to each beam by two *galvanized* iron screw bolts (*see* Chap. XV.); these are driven through the plank and beam flange, and have nuts upon their points. Fig. 91 shows these connections to an angle bulb beam in plan and elevation, from which it will be noticed that the heads of the bolts are sunk some distance below the surface of the plank, the holes above them

FIG. 91.



being filled with wooden plugs dipped in white lead to ensure the watertightness of the bolt holes. This sinking of the heads prevents their exposure until the deck is very considerably worn. The ends of the planks are supported by short pieces of plate which are riveted to the beams, and have a width equal to that of the planks. Fig. 92 shows this arrangement when the beams are of angle and tee bulb respectively.

FIG. 92.



Where a wood deck is laid over plating it is fastened to the latter between the beams, so as to stiffen the plating as much as possible.

Watertightness is secured at the edges and butts of the planks by caulking them, the oakum in the seams being afterwards covered with pitch.

CHAPTER X.

WATERTIGHT SUBDIVISION OF SHIPS.

As the thin skins of iron ships are more easily penetrable than the thicker bottoms of wooden vessels—especially as these latter usually had the spaces between their frames filled in solid with wood up to the turn of the bilges—considerations of safety early demanded the division of the former into watertight compartments (*see* page 14). This is especially necessary in war ships, since they are liable, in action, to under-water attacks from ram and torpedo, in addition to damage from accidental collisions, grounding, &c. The sub-division in these vessels has therefore been made more and more minute as the above forms of attack have been developed, so that it is now usual to divide the largest war ships into more than one hundred watertight compartments.

The required sub-division is attained by (1) fitting horizontal watertight iron or steel decks and platforms ; (2) working vertical *transverse* and *longitudinal* (fore-and-aft) bulkheads; and, in many ships, by (3) the adoption of an inner bottom, the double-bottom space being partitioned off into a number of watertight cells or compartments as described in Chapter VI. Plate X shows the subdivision of the large armourclad already described, the "plan of protective deck" indicating the bulkheads above that deck, whilst the "plan of hold" well shows how the spaces below the watertight platforms at the ends, and below the protective deck amidships, are divided. The main partitions are shown by thick lines and the smaller bulkheads by thin ones.

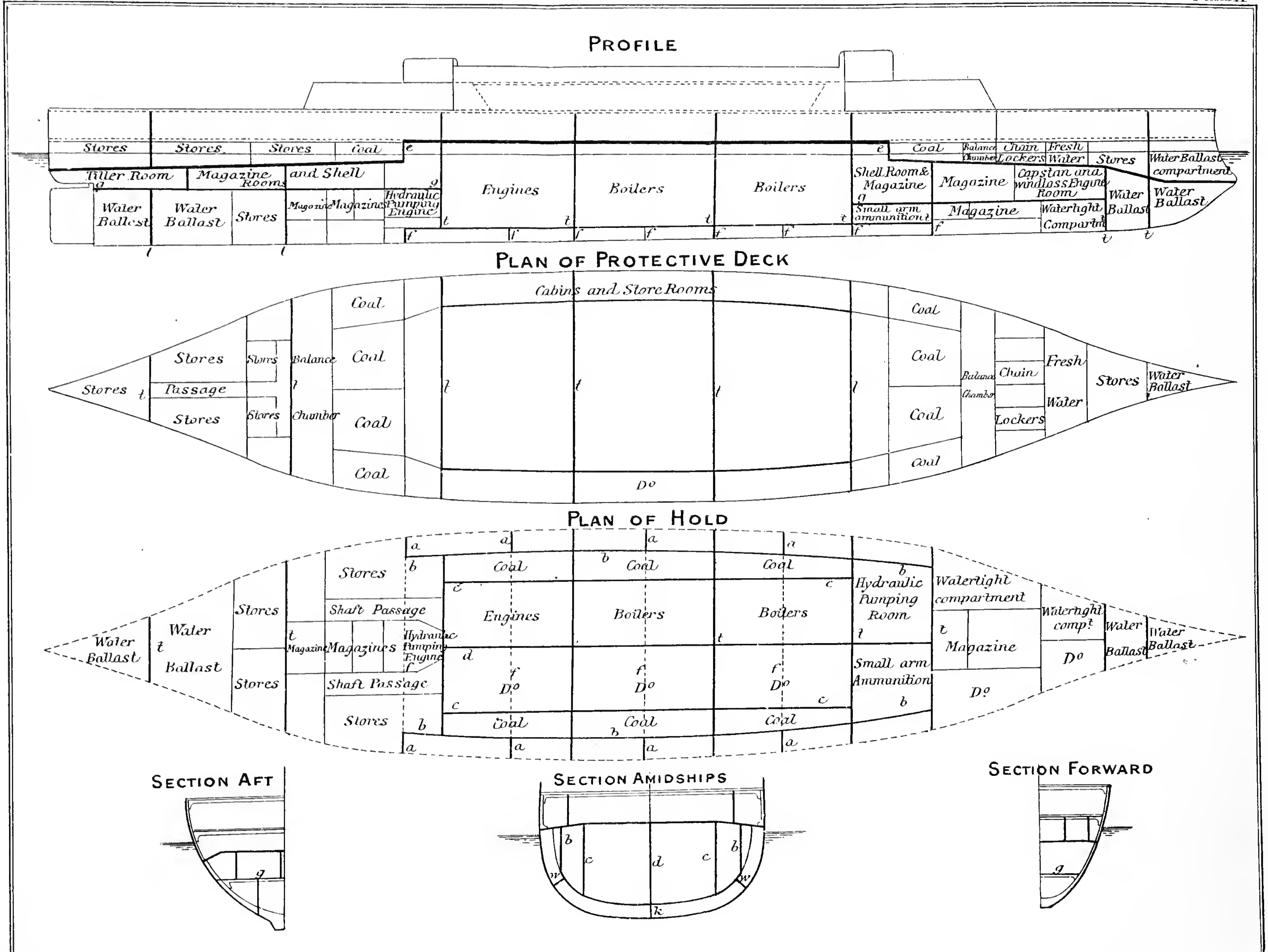
The main transverse bulkheads are marked *t* in profile and plans. The foremost one is known as the *collision* bulkhead ; that placed at the after end of the shaft passages is the *stuffing-box bulkhead*, which is made watertight where the propeller shafts pass through it by means of stuffing-boxes. Of the others, one is disposed right aft, and the remainder enclose the machinery, magazine, &c. spaces. Those in wake of the inner bottom have their lower edges bounded by it, and their sides above the longitudinal next below armour shelf on each side extend continuously to the outer bottom ; below those longitudinals each bulkhead is generally continued to the outer bottom by means of a watertight transverse frame. Those bulkheads which extend above the protective deck are necessarily divided into two parts by that deck.

The main longitudinal bulkheads are those forming the wing passages and coal bunkers,—marked *b* and *c* respectively in the sketches,—and the middle line bulkhead *d*. The last named divides the machinery for working one propeller from that actuating the other, so that in the event of one side of the vessel being broken through by a ram or torpedo, the bulkhead would restrict the damage to one screw only, an especially important advantage in ships provided with little or no sail power. In some vessels having twin screws this bulkhead is omitted, and one engine room is then placed abaft the other ; this secures the same isolation of machinery as before. For precautions regarding the height to which the main bulkheads, both transverse and longitudinal, should be carried above water, *see* page 17.

The horizontal watertight partitions comprise the upper deck (with the exception of the part between the barbettes, which is not plated from side to side) ; the protective deck worked over the side armour, this main deck being made watertight to the ends by thin plating marked *e e* in profile and shown clearly in section in Fig. 74 ; the protective lower deck at the ends ; and the watertight platforms *g g*, which virtually take the place of the inner bottom towards the extremities of the ship.

It will be noticed that the *wing spaces* above the watertight longitudinals (*w* in the section amidships) are divided into several parts by the transverse bulkheads ; and they are further subdivided by partitions marked *a* in the plan of hold ; the transverse watertight frames in double bottom are lettered *f* in the same plan (where not covered by a bulkhead) and in the profile. The entire space between the two skins is thus partitioned off into a large number of relatively small compartments. It may be added that many vessels in the mercantile marine are now provided with inner bottoms, principally with a view to utilising the double bottom space for water ballast purposes ; but of course they are also valuable as an additional security against accident.

In addition to the divisions already enumerated, the bulkheads enclosing store rooms, magazines, shaft passages, etc., are made watertight, thus rendering the sub-division still more minute. The total number of watertight compartments below the protective deck of the vessel under consideration is 110, of which 30 are in the double bottom spaces and wings. It will be seen from the profile, and plan of protective deck, that the spaces in the vicinity



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of the water line ("between wind and water") at the unarmoured ends are well sub-divided, as is also the part above the protective deck amidships, for the reasons given on pages 16, 44, and 47.

The largest compartments are those containing the engines and boilers, and if one of the boiler rooms is damaged by a ram or torpedo, so that the wing passage and coal bunker bulkheads in its vicinity are also injured, the total amount of water which would find its way into the ship, supposing the compartments empty, would be about 700 tons. Since, however, a considerable volume is occupied by coals and machinery, the quantity of water entering the vessel would be considerably less than given above, and the vessel will heel over a few degrees only. This heel can be lessened, if thought advisable, by voluntarily admitting water to the wing spaces and double bottom on the side opposite the damage. The capacity of one of the largest double-bottom compartments is about 40 tons.

The extent of the sub-division in several other armourclads is exhibited in the following table, taken from the *Manual of Naval Architecture* previously mentioned.

Ironclad Ships of Royal Navy.		Watertight Compartments.		
Classes.	Names.	In Hold Space.	In double bottom and wings.	Total.
Largest early types.	*Warrior* -	35	57	92
	Achilles -	40	66	106
	Minotaur -	40	49	89
Smaller early types.	*Hector* - -	41	52	93
	Resistance -	47	45	92
Largest recent masted types.	*Monarch* - -	33	40	73
	Hercules -	21	40	61
	Sultan - -	27	40	67
	Alexandra -	41	74	115
	Téméraire -	44	40	84
Smaller masted types.	*Invincible* - -	23	40	63
	Triumph -	26	40	66
Belted ships -	*Shannon* - -	44	32	76
	Nelson - -	83	16	99
Mastless or lightly rigged.	*Devastation* - -	68	36	104
	Dreadnought -	61	40	101
	Inflexible - -	89	46	135
Rams -	*Hotspur* -	26	32	58
	Rupert - -	40	40	80
Monitors - -	*Gorgon* -	19	20	39
	Glatton - -	37	60	97

Unarmoured vessels are also well sub-divided, the *Iris* having 61 compartments, the "C" class 77, and those of the *Buzzard* type 30. A general idea of the arrangements in the last two classes can be obtained from Figs. 52 and 53.

The value of efficient watertight sub-division has been abundantly demonstrated in actual experience, and one or two examples will suffice. In June 1880 the steamship *Anchoria* of the "Anchor" line was struck nearly amidships by the *Queen* and had 28 feet of water in the damaged compartment, but the bulkheads kept her afloat. As an illustration of the usefulness of the inner bottom it may be mentioned that the *Great Eastern*, whilst proceeding to New York, touched upon a rock and tore a large hole in her outer bottom; the inflow of water was, however, limited by the inner skin, and no inconvenience resulted, the vessel arriving at her destination in safety. The *Iron Duke* also damaged her outer skin by grounding, but the injury was confined to two double bottom compartments. The accidental ramming of the *Bellerophon* by the *Minotaur* caused part of the wing passage of the former to fill, but the water was prevented from flowing into the hold by the wing passage bulkhead. Finally, the cases of the *Canada* and *Hecla* may be cited as two recent examples showing the utility of the collision bulkhead. The first collided with a barque near Halifax, and the second with a steamship in the English Channel, but in each case the damage done was confined to the foremost compartment.

Attention will next be directed to the structural details of the bulkheads of the vessel shown in Plate X., those of smaller ships being constructed in a very similar manner. The plates are arranged in horizontal layers, and are from $\frac{1}{4}''$ to $\frac{3}{8}''$ thick according to the size of the bulkhead and their position in it, the lower plates in any one bulkhead being slightly thicker than the upper ones.

The largest transverse bulkheads are those forming the engine and boiler rooms, and their method of construction is shown by the part elevation and section given in Figs. 93 and 94, from which it will be seen that the plates are worked with *flush* edges and butts, the former being joined together by edge strips made of T bars, $4\frac{1}{2}'' \times 2\frac{1}{2}'' \times \frac{3}{8}''$, which also efficiently stiffen the bulkhead against buckling stresses; these are shown at *a*. On the opposite side, angle bars *b*, $3'' \times 3'' \times \frac{5}{16}''$, are worked vertically 4 feet apart, and these effectually co-operate with the edge strips in stiffening the bulkhead. This stiffness is often further augmented by fitting

Fig. 93.

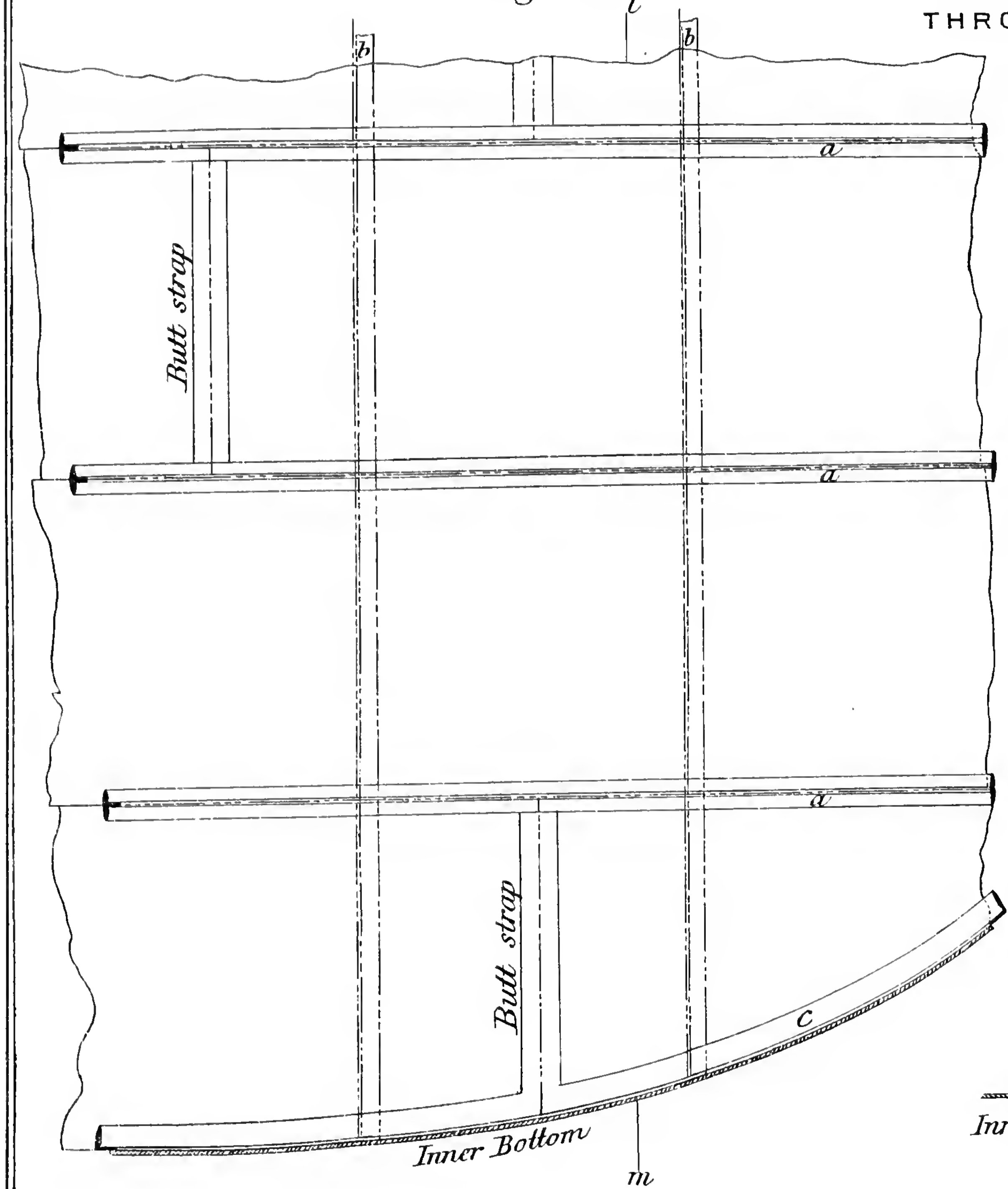


Fig. 94.
SECTION
THROUGH l.m.

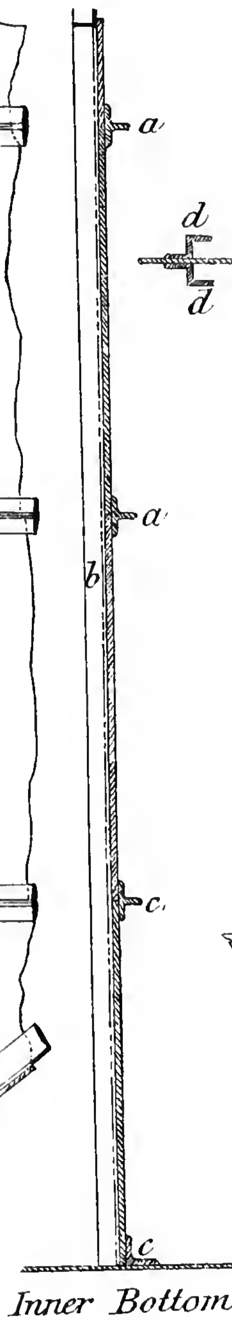


Fig. 95.

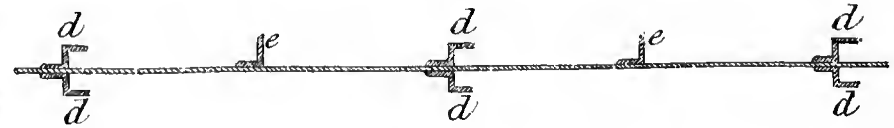
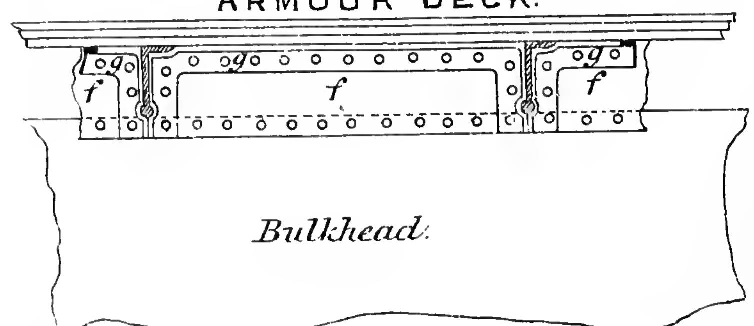


Fig. 96.

ARMOUR DECK.



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a strong 10-inch Z bar at intervals, instead of one of the vertical angle bars. The butt straps fit between the T bars and, with these edge strips, are single riveted. The bounding edges of the bulkhead are united to the inner bottom, protective deck, &c., by an angle bar $4'' \times 3\frac{1}{2}'' \times \frac{7}{16}''$ marked *c*.

The main transverse bulkheads worked towards the extremities of the ship have *lapped* edges and butts, and are stiffened by vertical angle bars only, spaced from 2 to $2\frac{1}{2}$ feet apart. These bulkheads often receive good horizontal stiffening by the abutment against them of the various decks and platforms; for, unless a deck is plated, the bulkhead passes continuously through it, horizontal angle bars being provided on the latter to support the ends of the wood deck. The bounding edges of these bulkheads are secured to the outer bottom plating by single angle bars, and care is taken to make the bulkheads watertight where the continuous portion of the longitudinals passes through them.

All the longitudinal bulkheads of this vessel are fitted between the transverse ones. That placed at the middle line of the ship has flush edges and butts, both edge strips and butt straps being single riveted. The stiffeners are all vertical, and are shown in the horizontal section through the bulkhead given in Fig. 95; they consist of small Z bars $3'' \times 2\frac{1}{2}'' \times 3'' \times \frac{7}{16}''$ placed back to back as shown at *d*, and of intermediate angle bars *e*. The former are 4 feet apart, and act as pillars to the deck above. Additional stiffness is given to this bulkhead also by omitting a few of the vertical angle bars and substituting 10″ Z bars. A single angle bar connects the edges of the bulkhead to the transverse ones between which it is worked, and to the inner bottom, whilst its upper edge is made watertight as indicated in Fig. 96. The bulkhead plates stop just beneath the deck beams, and are riveted to short pieces of plate *f* fitted between those beams; these latter plates are joined to the angle bars *g* which closely fit against, and are riveted to, the deck plating and beams, and which can then be caulked.

The wing passage and coal bunker bulkheads have their plates arranged on the raised and sunken strake system (*see* Plate III.), and their bounding edges are secured as just detailed for the middle-line bulkhead. They are stiffened by vertical angle bars 2 feet apart, the coal-bunker bulkheads having additional 10-inch Z bars at every 8 or 10 feet in lieu of the angle bars, to assist those bulkheads in resisting the pressure which would be caused by a torpedo bursting in contact with the outer bottom. The wing-passage bulkheads in some recent ships have been made thicker ($\frac{3}{4}''$) than usual for the same reason.

The smaller bulkheads such as those to the shaft passages, shell and provision rooms, &c., are $\frac{1}{4}''$ thick, and have vertical angle-bar stiffeners 2 feet apart.

Each bulkhead and watertight platform is tested for watertightness after it has been carefully caulked, either by filling the enclosed compartments with water, or by playing water upon the bulkheads from a hose. Any *weeping* of the rivets or caulking which results, is rectified, and the test again applied until perfect watertightness is secured.

It is evident that all openings cut in watertight partitions for convenience of access to the various compartments must be capable of being made watertight. These openings include the *manholes* cut in the inner bottom for access to the double-bottom spaces, the *scuttles* in the watertight platforms, and the doorways in the main bulkheads, as well as those to the various store rooms. Fig. 97 shows the method adopted for making a scuttle watertight. An angle bar *m* is riveted round the aperture, and to it the steel cover is fixed by the hinges *h*. On the under side of this cover, round its edges, a strip of india-rubber is worked, this being kept in place by thin strips of iron which run along its edges and are screwed to the cover. This is clearly indicated in the enlarged section through A B. To secure watertightness the india-rubber is forced tightly against the angle bar *m* by means of *butterfly nuts*, the screws of which are hinged to *m* and turn up into recesses cut in the ends of the hinges and in *lugs l* riveted to the cover ; the nuts are hove down until the india-rubber is well compressed. To lift the cover, the butterfly nuts are slacked back and the screws turned out of their recesses as shown at *d, d* ; the other nuts in the figure are indicated in their securing positions.

Manholes are made watertight in a manner similar to that just described, except that the covers are sometimes *portable* instead of being hinged, and are oval in order to conform to the shape of the hole. Small screw plugs are fitted in manhole and watertight scuttle covers, the removal of which allows air to escape when the compartment below is being filled with water ; they can be easily replaced when the operation is completed.

Large doorways are generally cut to afford communication between the boiler-rooms, &c., as well as in the main bulkheads at other parts of the ships low down in the hold ; and since, in

Fig. 97.

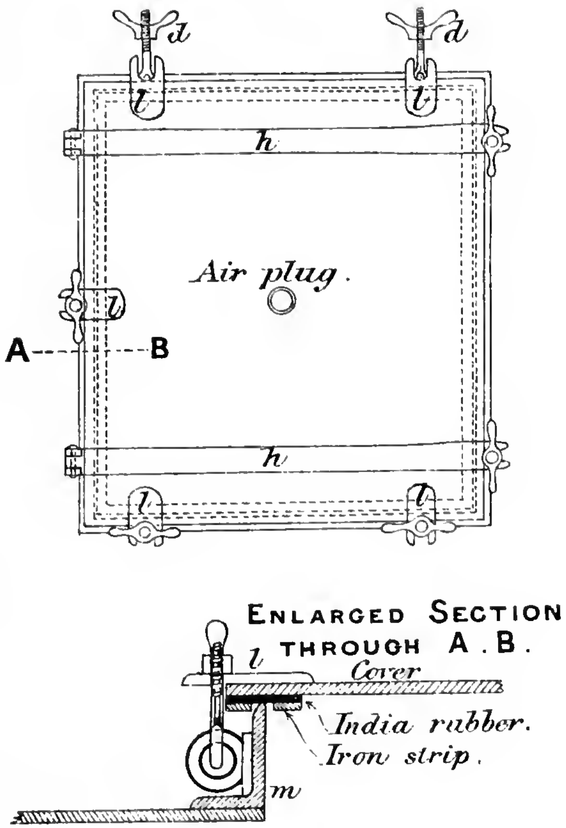


Fig. 98.

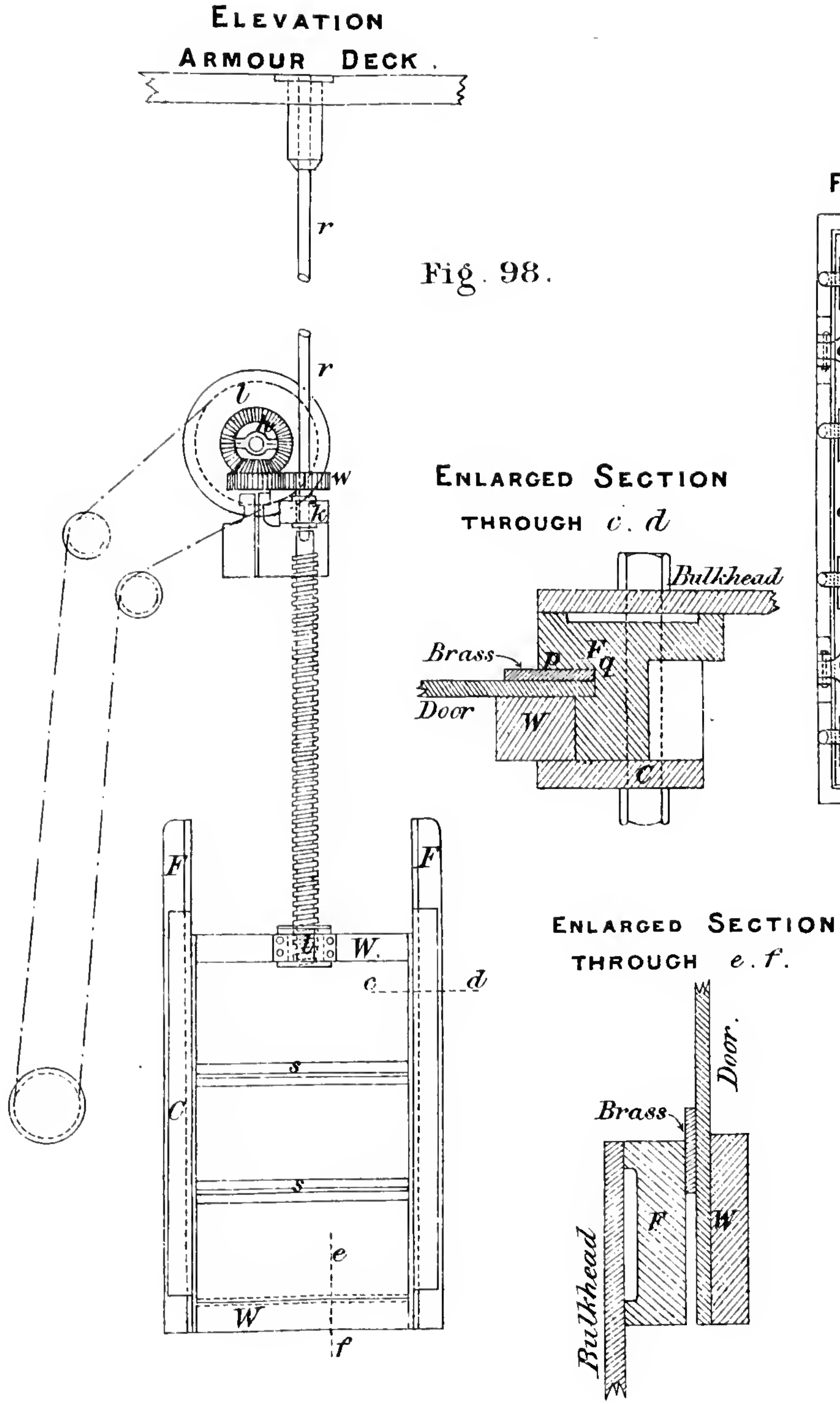
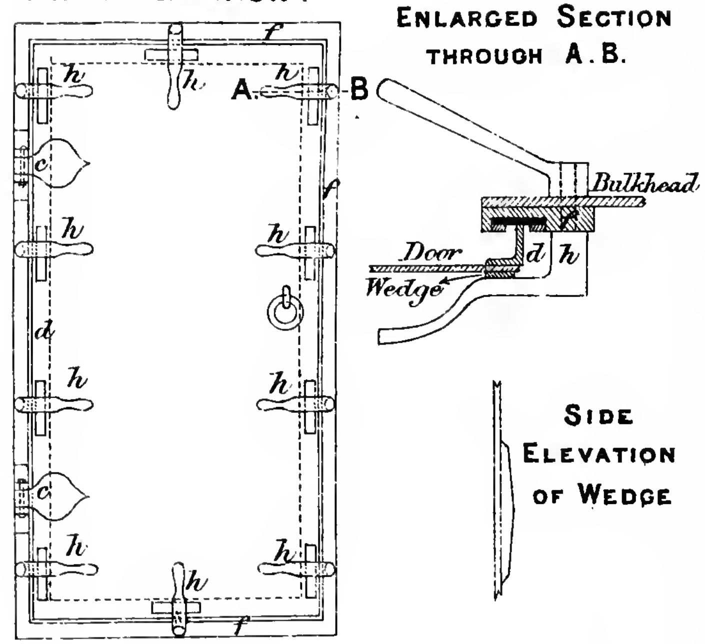


Fig. 99.



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the event of a large breach being made in the side, the water would probably rise too rapidly in the compartment to admit of the doors being closed from below, if previously open, provision is made for shutting them from the first deck above water. These doors *slide* in either a horizontal or vertical direction, the fittings for one of the latter being given in Fig. 98. From the elevation, and enlarged sections through *c d* and *e f*, it will be seen that a frame F is worked all round the doorway, its sides, of the section shown, tapering from top to bottom so that from the thickness indicated in the section through *c d*, each side becomes reduced in thickness to that of the bottom part of the frame, (shown in the section through *e f*) where it joins that part. The part *p q* of each side is at the same distance from the bulkhead throughout the length of the frame, and thus each forms, with a *cheek* C (bolted to the frame and bulkhead), a wedge-shaped groove in which the door works. This door consists of a wrought-iron plate, stiffened by T bars *s*, *s*, and secured at its edges to a wrought - iron frame W, the sides of which are tapered to the same extent as those of the frame F. Round the edges of the door, on the side opposite W, strips of brass are screwed as indicated in the sections, and thus, as the door descends in the grooves, these strips are forced against the frame F by the wedge-action of the door, watertightness being thereby secured. The door is actuated by a rod *r*, the upper end of which terminates at the first deck above water, whilst its lower end is jointed to a screw S, which works in the nut *t* attached to the upper part of the door. The rod is made square at its upper end, so that it may be turned by a spanner made to fit over its head. As the rod is prevented from moving vertically by the bearing *k*, the door must be raised or lowered as the screw revolves, the latter movement occurring when the spanner is turned with a right-handed motion. A *deck socket* is fitted to receive the upper end of the rod, and contains a tell-tale arrangement worked by the rod, which shows whether the door is open or shut. Fig. 98 also indicates the method of moving the same door from below. A spindle passes through the bulkhead and carries the pulley *l*, and the bevel-wheel *h*; round the former, as well as round other wheels attached to the bulkhead, a chain passes as shown. The rod *r* is fitted with a cog wheel *w*, which, together with *h*, gears with a third wheel. Turning the pulley *l* by means of the chain thus causes S to revolve and work the door. A second pulley is often fitted to the spindle on the opposite side of the bulkhead so that the door may be actuated from either side.

In *horizontal* sliding watertight doors, the frame round the doorway is placed horizontally, and the vertical rod for moving the door has its lower end in a bearing fixed to this frame on its lower edge. The rod carries two cog wheels or *pinions* at different parts of its length, and each gears into a corresponding *rack* fixed to the door, so that the latter is moved as the rod turns. The bottom groove in the frame is liable to become choked by dirt, which would interfere with the closing of the door; to prevent this the groove is covered by a sill which is automatically removed by the door when being closed.

When magazines, store rooms, &c., are not in use, their entrances are kept closed, *hinged* (or swinging) watertight doors being employed for this purpose; similar doors are also used for those openings in main bulkheads which are situated high up, as there would be ample time to close them in the event of an accident. The method of constructing one of these doors can be seen from Fig. 99. A frame *f* surrounds the doorway and has a groove cut along its sides and ends, into which strips of india-rubber are received; these are secured at their edges by narrow thin pieces of iron as indicated in the section through A B. The door consists of a thin wrought-iron plate with an angle bar *d* riveted round its edge, and is hinged at *c* to the frame of the doorway. To fasten the door in place, several handles marked *h* are employed. These turn in sockets fitted in *f*, and are so shaped as to press against vertical brass wedges on the door, when the latter is in place. Thus, the harder the handles are forced upon these wedges, the more will the india-rubber be compressed by the angle bar upon the door, and it is this compression which ensures watertightness. It will be noticed that the door may be secured from either side of the bulkhead, and also by turning the handles in either direction, since the wedges are tapered at each end, as will be seen from the side elevation of one of these given in Fig. 99. To allow of that edge of the door to which the hinges are attached being well forced against the india-rubber, the bolt hole in each hinge is made oval, so that the door is not rigidly bound at those hinges.

It will be obvious that in the time of trial the value of the bulkheads will depend upon the rapidity with which the doors in them can be efficiently closed, and hence the necessity for frequent drill and inspection to see that they are in thorough working order. In the Royal Navy this inspection takes place weekly.

CHAPTER XI.

PUMPING, FLOODING, AND DRAINAGE ARRANGEMENTS.

The above arrangements provide for :—

(1). Voluntarily admitting water into the ship, either to alter her trim or heel, to increase the stability, or to fill the magazines, spirit rooms, &c., in the event of their being threatened by fire. These operations come under the head of *flooding*.

(2). Clearing water out of the ship which has been allowed to enter voluntarily ; or which has accumulated through ordinary small leakages, by accident, or from damage done in action. Also for distributing salt water throughout the ship for wash deck, sanitary, and fire purposes, as well as for conveying fresh water from the storage tanks to the filters, galleys, &c. For these purposes pumps driven by steam, as well as smaller ones worked by hand, are used.

(3). Collecting water which is to be pumped overboard into convenient places for this purpose. This is done by a number of pipes, which form the *drainage* system of the vessel.

It will be noticed that the pumps are not relied upon to eject water from a ship as fast as it enters through a large leak, the inadequacy for such a task of the pumping power usually supplied to a vessel being easily demonstrated. If a hole is situated h feet below water, the velocity with which water enters the ship through it is given by the following rule :—

$$\text{Velocity} = \sqrt{h} \text{ feet per second.}$$

Hence, if A represents the area of the hole in square feet, the amount of water passing through it in one second is $8 A \sqrt{h}$ cubic feet.

The maximum quantity of water which can be thrown overboard in an hour by the armourclad most liberally supplied with pumps, does not exceed 4,500 tons, or $1\frac{1}{4}$ tons per second ; and the above rule will enable an estimate to be made as to how large a leak, situated, say, 9 feet below water, can be just kept under by these pumps. If A represents the required area of hole, the quantity of water entering the ship in each second is $8 A \sqrt{9} = 24 A$ cubic feet. The amount of water got rid of by the pumps is $1\frac{1}{4} \times 35$ cubic feet. Hence

$$24 A = 1\tfrac{1}{4} \times 35.$$
$$\text{or } A = 1{\cdot}82 \text{ square feet.}$$

Remembering that a successful ram or torpedo attack would probably result in making a hole some 20 or 30 square feet in area, it is seen how hopeless would be the case of a vessel when so attacked, if dependent upon the pumps alone for safety.

This calculation shows how necessary is the watertight subdivision described in the preceding chapter, care being taken in time of accident to isolate the damaged compartment from the others. As that compartment fills, the rate of inflow of the water lessens, and then a leak stopper of some description may be successfully got into position over the hole. If this can be accomplished, the pumps can be set to work to empty the compartment so that any possible temporary repairs may be effected from inside.

STEAM PUMPS.—In arranging the steam pumping power of a ship the *circulating pumps*, designed for use in connection with the condensers of the main engines, are fitted so that they may be employed in case of necessity for pumping water out of the ship, this water being then utilized for condensing purposes instead of drawing from the sea. For this reason the pumps are fitted with *bilge suctions* (pipes with their ends opening into the bilges, where the water to be pumped overboard is accumulated) through which water from the ship can be drawn into the condensers before it is discharged overboard. These pumps are now driven by independent sets of engines, this being preferable to the earlier plan of working them from the main engines, because, in the latter method, any reduction in the number of revolutions of the propelling engines consequent upon an accident, correspondingly diminishes the power of the pumps. Again, when independent engines are used they can be placed high up in the engine room and considerably above the pumps worked by them ; the latter can therefore be kept going even when the engine room is partially flooded. The *air pumps* in some vessels have also been utilized for the same purpose. The other steam pumps ordinarily fitted in large ships are, the *main engine bilge pumps* worked by the propelling engines and used for pumping moderate quantities of water from the bilges ; the *main* and *auxiliary fire and bilge* pumps, actuated by separate engines, and used for fire and wash deck purposes as well as for pumping from the bilges ; small "hand" pumps fitted so as to work from the main engines or by hand ; and, in some modern vessels, *Friedmann's ejectors*. The last are supplied in order to discharge rapidly overboard a large quantity of water, especially from the coal spaces at the extremities of central citadel ships, their comparative immunity from choking rendering them very suitable for that purpose. Each ejector consists of several nozzles with their axes on the same straight line, a small space separating each nozzle from the adjacent ones. To one end of the ejector a *steam pipe* from a boiler is led, whilst a larger *discharge* pipe is connected to the other. The ejector is surrounded by a strainer and placed in an *ejector tank*, in which is collected the water to be discharged overboard. On steam from the boiler being forced through the nozzles a partial vacuum is created, this being filled by water passing between the nozzles, which water is then forced overboard through the discharge pipe by the pressure of the steam. An ejector having a $4\frac{1}{2}''$ steam pipe and an $8''$ discharge pipe is said to be capable of throwing overboard 300 tons of water per hour. They are, however, very wasteful of steam and have not been fitted in the most recent ships.

HAND PUMPS.—For a long time the largest pumps worked by hand alone were those known as *Downton* pumps. They usually had three buckets working in the pump barrel, with a valve in each, and their principle of action was exactly the same as that of the common lift pump; but, by having three buckets the flow of water was made continuous and they were much more powerful than one of the common pumps with the same lift. Those now used, however, are much simpler in construction, and doors on the outside of the pump enable the valves to be easily got at and cleared should they become choked from any cause. A skeleton sketch of one of the latest pumps used is given in Fig. 100. A solid watertight piston P is rigidly attached by a rod to a slide S, which is made to work up and down in the head of the pump by means of a crank (shown in side elevation) which passes through the pump, and can be turned by manual labour; this operation causes P to move up and down in the pump barrel. In that barrel an orifice *e* is cut at its lower extremity, whilst a similar one *f* is formed at its upper part. The water to be pumped overboard enters through A, and its passage to the pump barrel is controlled by the valves *a*, *b*, *c*, *d*. Consider the action of this pump during one complete stroke, the piston being initially in its lowest position, as shown in the Fig., and the pump barrel being assumed full of water. As P rises, the water above it is lifted and forces its way through the valve *d*, and thence escapes through the delivery pipe D. But as the piston rises, a vacuum is formed behind it, which causes water to pass into the pump barrel through A, the valve *a* and the orifice *e*. When therefore P has reached the top of its stroke, the pump barrel is again full of water which, when the piston descends, is forced out of the valve *b* and thence to the delivery D. But, in descending, the piston creates a vacuum above it, in consequence of which water flows through A, the passage *g*, the valve *c* and the orifice *f*, so that at the end of the stroke the pump is exactly in the same condition as at the beginning, and the pump barrel has been completely emptied twice. It will be noticed that the piston of this pump is effective in both the up and down stroke, acting alternately as a lift and force pump; it is thus superior to the Downton, which —having valves in its buckets—is effective during the up strokes only.

The largest sized Downton pump ordinarily used was 9″ in diameter, and was estimated to discharge 90 gallons of water per minute or 25 tons per hour; this is also the power of one of the 9″ pumps of the newer pattern illustrated in Fig. 100.

In addition to the foregoing, smaller *lift* and *force* pumps are also supplied for use in connection with the galleys, wash places, &c.

A large pump is made capable of drawing from several compartments in one of the two ways illustrated by Figs. 101 and 102. In the former, a suction-box or *valve chest* V is fitted beneath the pump and connected to the bottom thereof by the *tail pipe* shown. Pipes marked *a* lead from the several compartments into the valve chest, and their upper ends are fitted with *screw-down* valves, which can be opened and closed by the handles *h*; the details of these valves will be given a little later. All that is therefore necessary to put any particular compartment in communication with the pump is to lift up the valve which is over the end of the pipe from that compartment, the other valves in the chest remaining closed.

The second method of attaining the same result, shown in Fig. 102, consists in employing a *deck-* or *suction-plate* D, to the under side of which, at its centre, the tail pipe from the pump is attached. The upper ends of the pipes from the several compartments are each led underneath the plate to a separate hole in the plate D, near its edge, and are usually kept covered by screwed caps marked *b*. On the upper side of the plate a *goose-neck* or arched pipe *g* is fastened at the centre hole, around which it can revolve; and its length is sufficient to enable it to be screwed at its other end to any of the suctions, the covers *b* being removed for that purpose. To enable the pump to draw through any particular pipe, the cover from that pipe is taken off and the goose-neck attached in its place, as indicated in the figure. When a deck-plate is adopted, it should always be well above the water-line, to prevent the possibility of water entering the ship through a

Fig 100.

ELEVATION OF PUMP

SIDE ELEVATION OF CRANK

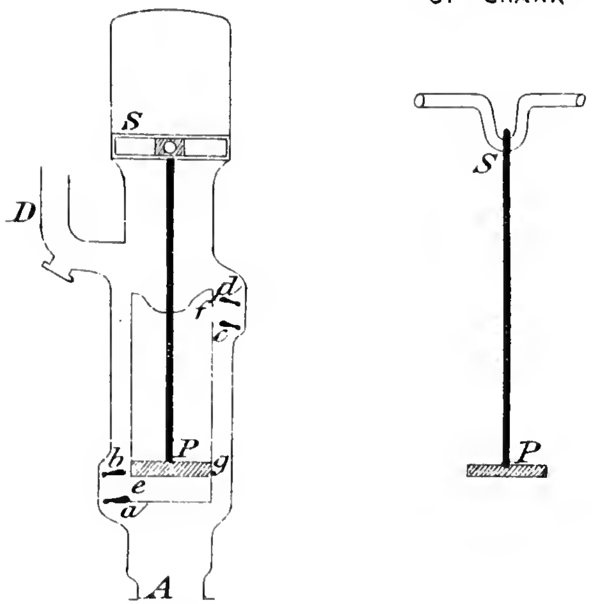


Fig. 101.

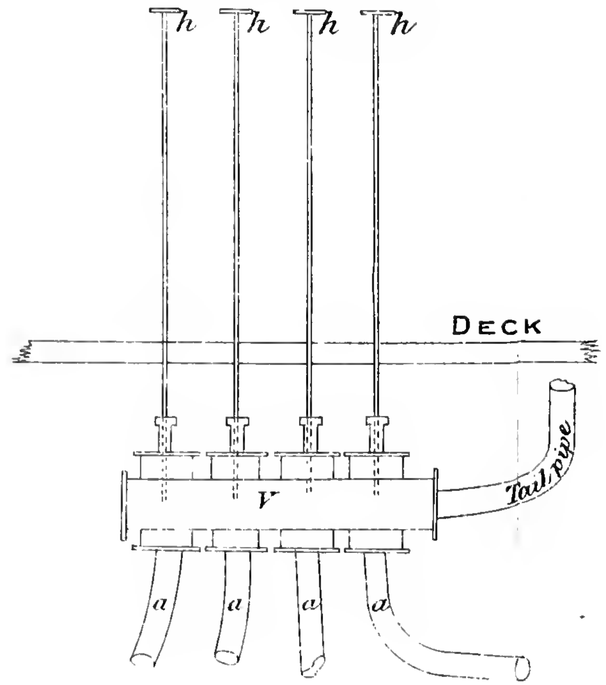


Fig. 102.

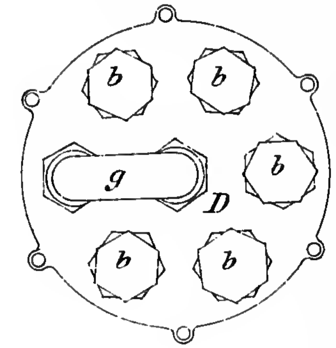


Fig. 104.

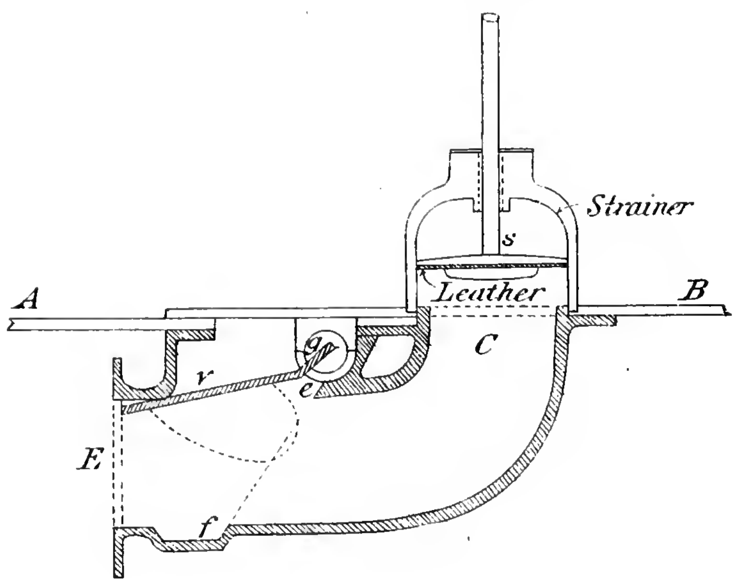


Fig. 105.

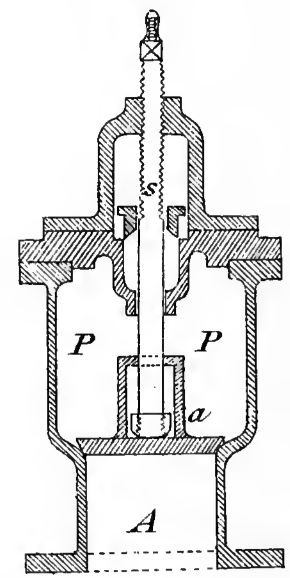
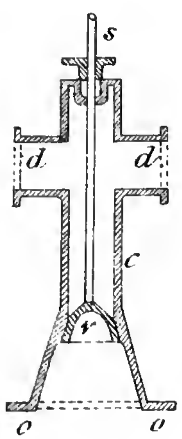


Fig. 103.



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pipe leading from a damaged compartment, or through the *sea suction* which is led from the side of the ship to the deck-plate in order to supply the pump with salt water; for this reason it is sometimes necessary to put the plate upon a pedestal.

Some of the valves largely employed in pumping and drainage arrangements will now be described. Each of the large hand-pumps has a sea suction leading from a *Kingston valve* in the side of the ship below water to the valve-chest or deck-plate of the pump. A sectional elevation through a Kingston valve is given at Fig. 103; it consists of a gun-metal casting *c*, secured to the inner surface of the ship by the flange *o*, and is usually made with two branches *d*, *d*. A valve *v* fits into its outer end, and is worked by a spindle *s* and a series of rods and levers from a deck above the water line; water enters when *v* is thrust outwards from the position shown. The sea suction of the pump is connected to one of the branches *d*, and is provided with a sea-cock close to the Kingston as an additional security in the event of the latter failing to act. The other branch *d* is used for flooding purposes, a pipe being lead therefrom to the compartment it is required to flood.

A common form of *non-return* valve for drainage purposes is given in section in Fig. 104. Its object is to permit the flow of water in one direction, viz., from above the flat A B, through C, and thence out at E, whilst preventing its passage in the opposite direction; and it is attained by fitting a valve *v*, hinged at *g*. This allows water to pass in the right direction, but if the water evinces any tendency to flow from E towards C, the valve falls by its own weight, and is pressed by the water against its seating *e f*, thereby closing the passage. A thin hollow copper float is somtimes attached to the back of the valve, as shown by dotted lines, so that the latter may be more easily lifted by the water. It will be observed that the orifice C is surrounded by a strainer to prevent the entry of any rubbish which might choke the valve casing. The orifice can be closed by a *screw-down* valve *s*, worked from above the water line, so that should the valve *v* get out of order, water may be prevented from passing upwards through C by closing the valve *s*.

Fig. 105 shows in vertical section a valve intended not only to act as a combined screw-down and non-return valve, but also to permit water to pass it in *either* direction when required. The valve *a* closes the passage A, and is controlled by the spindle *s*, this latter having a screw cut in its upper part, so that as the spindle is turned by means of a handle which can be fixed to its upper end, it moves upwards or downwards. To the lower end of s a nut is screwed, and when this bears against the bottom part of a, (as in the figure), the latter is acting as a screw-down valve, closing the passage A. Since the spindle passes freely through the upper part of the valve (which is at some distance above the nut), the former can be raised considerably without lifting the valve. This then acts as a non-return, allowing water to pass upwards through A, along the passage P, and thence out of an orifice not shown, but not permitting it to flow in the opposite direction. If, however, s be raised still more, the valve will be lifted from its seating, and water can then pass from P downwards through A. Had the upper part of the valve been made shorter, so that no movement of the spindle was possible without raising or lowering a, a simple *screw-down* valve would have been produced, to which those adopted in the valve-chest V of Fig. 101 are similar.

Having described the pumps and valves used, the way in which they have been utilized in a modern large war ship will now be pointed out, general sketches of these arrangements being given on Plate XIV. The steam pumps employed in this particular ship, and their capacity in tons per hour, are given in the following table :—

2 Circulating pumps	1200	tons
2 Main fire engines	150	,,
2 Auxiliary fire and bilge . . .	150	,,
4 Main engine bilge pumps . .	120	,,
2 " Hand " pumps	14	,,
4 Friedmann's ejectors	1200	,,
Total steam pumping power .	2834	,,

Of these, only the fire engines E and the ejectors with their tanks T are shown in the Plate. It will be noticed that the circulating pumps of the above vessel furnish a large proportion of the pumping power, but this is much more emphatically the case in the most recent vessels, where the ejectors have been omitted and the circulating pumps made more powerful ; some recent armourclads have *four* circulating pumps, each capable of discharging 800 tons of water per hour There are four 9-inch Downton pumps in the vessel under notice, one in each of the forward boiler rooms and one in each engine room, each pump being fitted into a *pump room* cut out of the coal bunker for this purpose, as shown in the sections of the vessel taken through the pumps. The forward ones are marked A, and those aft B ; and they are worked by cranks or handles H fixed above the main deck, from which deck their Kingston valves K are also controlled. Two smaller Downton pumps on the main deck are used for the fresh-water service, whilst eight small *lift* and *force* pumps distribute salt water to galleys, baths, &c.

For clearing water from the main compartments above the inner bottom a *main drain* D D, 12 inches in diameter, is worked through the double bottom on each side of the middle line, two cisterns C being fitted in each to receive the water. Small quantities of water in the engine and boiler rooms are conveyed to the main drains through short pipes fitted with combined screw-down and non-return valves S similar to those illustrated by Fig. 104; but for dealing with considerable quantities of water in those rooms, a large pipe is led from each compartment to the main drain beneath it, having its inner end covered by a *sluice valve* V (virtually a small watertight door), worked horizontally by means of rods leading above the water line. Water collecting above the watertight longitudinals finds its way to the main drains through small branch pipes as shown in section 1; and branches also lead to the main drain from the compartments near the ends of the ship.

To pump out the water collected by the drain pipes, suctions are led to one or the other of the cisterns C from each of the steam pumps, except the ejectors, and from the 9-inch Downton's. Thus all these pumps can be engaged simultaneously if necessary. The circulating pumps are only used when getting rid of large quantities of water; and in later vessels they are not connected to the main drain, but have the lower ends of their bilge suctions placed a little above the inner bottom of the engine rooms, the water being conducted to them from the other main compartments along the inner bottom, either through vertical sluice valves on the main transverse bulkheads, or through drain pipes above the inner bottom leading from those compartments. The main drains are connected by short pipes having stop valves in them, so that the pumps connected with one drain can be made capable of taking water from the other if required. A suction is also led from one or other of the Downton's to each compartment beyond the double bottom.

To empty the double bottom spaces, a suction known as a *stand pipe* is led from each compartment to a valve chest, the lower end of each pipe being protected by a drain box or strainer; several of these suctions are led to the same valve chest, as indicated at *a*, *b*, *c* and *d* respectively. These latter are similar to the one illustrated by

Fig. 101 and are each connected to the suction box of a Downton, the joining pipe containing a screw-down non-return valve (*w* in section 1) which works on the same principle as that shown in Fig. 105; this valve is ordinarily in its "non-return" position. As an example of the manner in which a compartment of double bottom can be drained, suppose it is required to empty a compartment in connection with *a*; the valve over the pipe leading from the pump valve chest to *a* is opened, as also the one in *a* which closes the suction from the compartment in question. Direct communication is thus established between that compartment and the pump, and the water is thrown overboard through a discharge pipe *x* leading into a scupper. It will be noticed that *a* and *b*, as also *c* and *d*, are joined together, so that if the pump on one side should break down, its work may be performed by that on the other side by opening the valve *f*.

To get rid of large quantities of water from the coal and other spaces above the protective deck forward and aft (where water is likely to accumulate from damage in action), several large drain valves *h* are fitted to open from beneath the deck; from these the water is conducted by pipes to the ejector tanks, and is then thrown overboard through the discharge pipes *g*.

The flooding arrangements will next be noticed. They are comparatively simple, for, since the compartments to be filled are below water, they are in most cases flooded by opening a pipe leading directly to each from a Kingston valve. The following are the modes of filling several specimen compartments of the vessel under consideration. To fill the water ballast chambers forward, a flooding pipe is led from the Kingston of one of the pumps A to the point *l* in profile. By this both ballast compartments forward may be filled if the sluice valve *m* is opened; and may be emptied by opening the sluices *m* and *n* so as to admit of the water being pumped out by the suction *o* from one of the pumps, or after being led into the main drain through the valve *p*. Similar methods obtain for flooding the ballast compartments aft, and the magazines. In the latter case a special flooding valve is fitted as near the magazine as possible, which can be opened from a deck above water; when this deck is closed in, the valve is also made to open from below water near the magazine, whence it can be operated in the event of the rod leading to the deck above being damaged, or inaccessible on account of fire. No special arrangements are made to pump out the water which would be used in flooding magazines, since it could be easily got rid of by passing into it an ordinary suction hose. Any particular double bottom compartment can be flooded through one of the Downton pumps. Suppose, for example, it is necessary to fill a compartment in communication with *a;* the various valves are opened as for pumping from that compartment, and *w* is, in addition, raised to its fully open or "flooding" position, so that water can pass it downwards. Then, when the sea suction of the pump is opened, water will flow into the pump suction box and thence into the compartment. In some vessels these spaces have been flooded by leading an independent pipe from a Kingston to the *standpipe* valve chest; and in others, where the deck plate has been adopted, they have been filled in the following manner. A flexible hose, known as a *dale hose*, is led from the delivery nozzle of the pump to the pipe in the deck plate communicating with the compartment to be flooded, whilst the gooseneck is attached to the sea suction. After a few strokes of the pump, water is drawn from the sea and delivered into the compartment, and since the pump then becomes a syphon, the flow of water continues without further pumping.

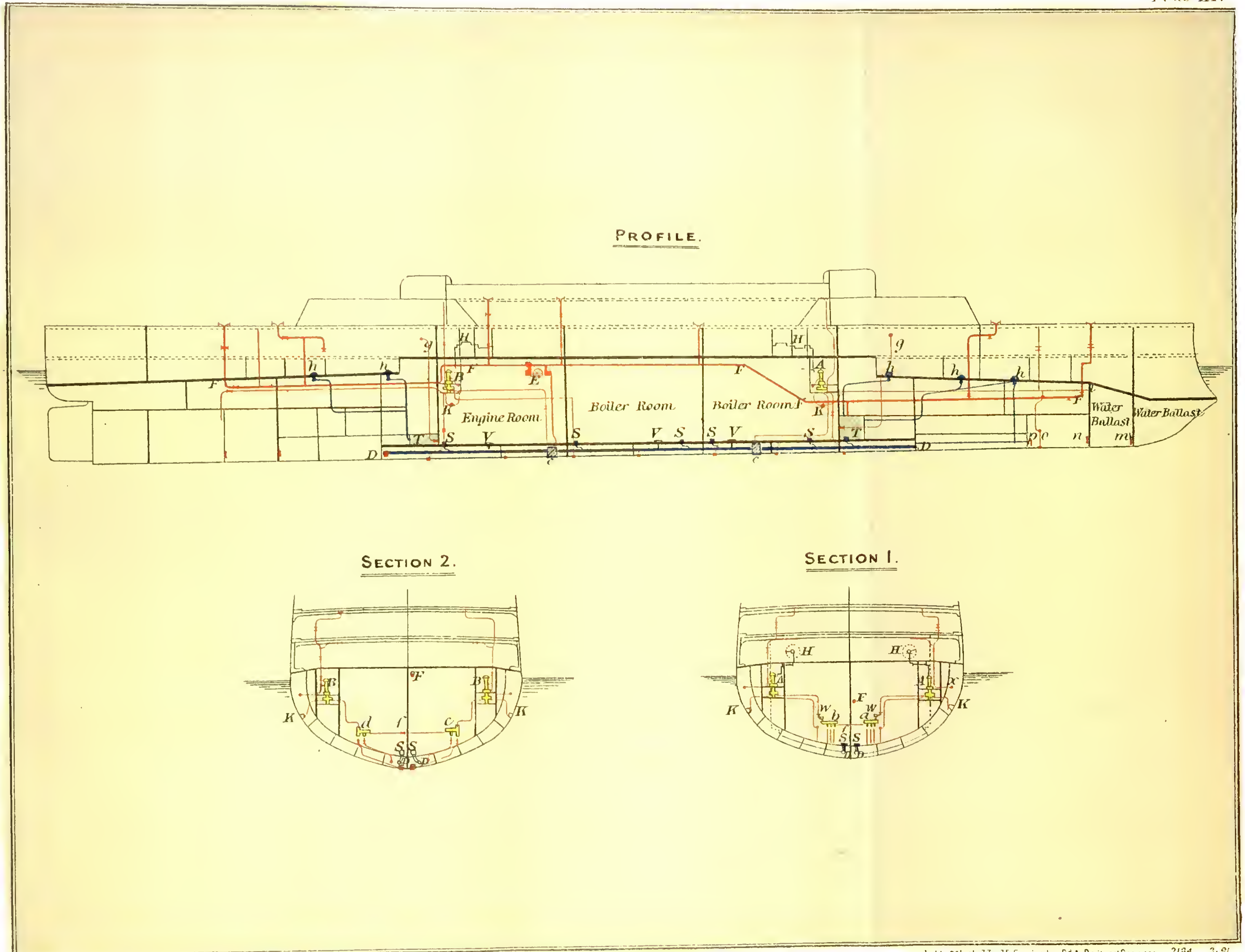
Means are provided to allow of the escape of air from a compartment when it is being flooded. One example of this will be remembered in the covers of hatchways formed in watertight platforms, which have air escape plugs as mentioned in the preceding chapter; and similar plugs are also fitted in watertight longitudinals. For the same purpose small pipes are led from other watertight spaces, such as water ballast compartments and wing passages, opening well above the water line. Magazines are provided with special air escape valves at their crowns, or the ventilation pipes are utilised for the same purpose.

Provision is made against fire by working beneath the protective deck a *fire main* F F, 5 inches in diameter, into which the fire, bilge, and Downton pumps can force water drawn from the sea. From this main several *rising mains* pass vertically through the decks as shown, having nozzles between these decks and at their upper ends, the latter being portable. Hoses can be attached to these nozzles, and a good supply of water thus obtained at any part of the ship for fire or wash deck purposes. Rising mains for the same purposes also lead from the delivery pipes of the 9-inch Downton's.

The fresh water is distributed throughout the ship by an independent system of pipes. The two $5\frac{1}{2}$-inch Downton's employed have suctions from the fresh-water tanks and Normandy's condensers, and supply the filter tanks, officers' and seamen's galleys, hydraulic pumping engines, etc. These pumps, together with their copper pipes, are tinned on the inside.

The lift and force pumps draw from the sea-suctions of the various 9-inch Downton's, to supply the baths, wash places, galleys, etc.

Similar, but less elaborate, arrangements to those described above are also fitted in the smaller war vessels of the Navy.



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CHAPTER XII.

VENTILATION OF SHIPS.

VENTILATION consists in drawing or exhausting foul air out of the ship and providing in its stead a good supply of fresh air. This is necessary for the health of the crew, for the preservation of stores, and for getting rid of dangerous gases which would otherwise accumulate in the coal bunkers, besides being required to secure a good air supply to the boilers.

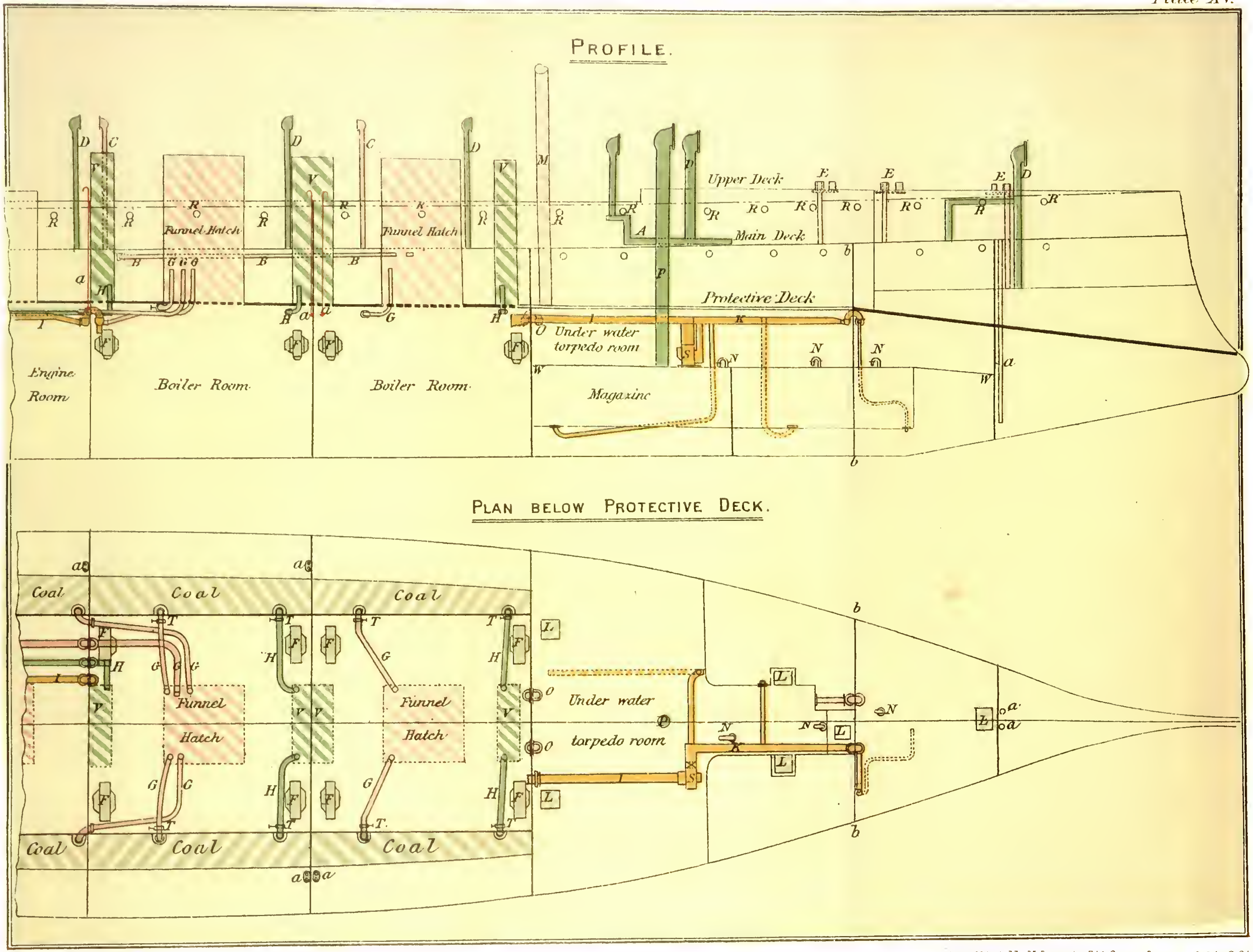
When a ship has a protective deck from end to end,—the number of openings in which is kept as limited as possible—it is usual, particularly in the larger ships, to supply air to the compartments below that deck by *artificial* means; that is to say, fans are employed to draw air from above the deck and to distribute it through a system of thin iron or steel pipes to the several compartments. In other cases, *natural* ventilation is principally resorted to, the supply of fresh air being obtained through hatchways, ports, &c., or through pipes fitted with hoods or *cowls* at their upper ends, and leading from above the upper deck to the compartments requiring ventilation.

The foul air is conducted out of the ship through *exhaust* pipes opening above the upper deck, or through a system of trunks and pipes put into connection with the funnel casing, the heat producing an up draught.

Plate XV. shows in profile and plan the ventilation of a vessel possessing a complete protective deck, the arrangements at the fore end only being given, and will alone be specially described, since they are practically the same at each end. The spaces between the upper and main decks get a plentiful supply of fresh air, principally through the hatchways in the upper deck, the scuttles R in the side of the ship, &c.; and vitiated air makes its exit through the same openings. A few of the smaller compartments between these decks get their fresh air as follows. A shaft conveying fresh air is connected to a horizontal trunk fitted through the compartments just above the main deck, as at A in the profile, from which trunk the air can enter into the various compartments through sliding shutters or *louvres.*

The compartments between the main and protective decks are principally supplied with fresh air through downcast shafts D provided with cowls at their upper extremities, so that as the ship goes ahead air is forced into its interior. The cowls can be turned round to face in any position, and thus when the vessel is at rest they may be turned to face the wind, in whatever direction it happens to be blowing. No cowls are, however, needed on those pipes which supply the upper coal bunkers, fitted above the protective deck, with fresh air, because the necessary down current is produced by the method which is adopted for getting rid of the foul air from those spaces. A trunk B B (*see* profile) leads from the bunkers into the funnel casing, and the up draught produced by the heat carries off the foul air, the place of which is supplied by fresh air flowing through the downcast shafts. The other spaces between the main and protective decks are either exhausted through the hatchways or through tubes C having their cowls turned away from the wind; or advantage is taken of the hollow towing bollards E and the mast M to utilise these also as uptakes, tubes being fitted to some of the former to connect them with the compartments requiring ventilation, and an exhaust hole cut in the latter just below the main deck. It should be mentioned that the cowls of the upcasts C can be turned *towards* the wind to supply the compartments below with fresh air if necessary; and the downcasts D can be similarly converted into exhaust pipes if required.

Of the compartments below the protective deck, the boiler rooms are well furnished with fresh air through the large ventilation shafts V opening well above the upper deck; it is down these ventilators that air is drawn by the steam fans F to supply the boilers when they are being worked under forced draught, and the engine rooms are also supplied with fresh air through similar shafts. To abstract the gases accumulating in the coal bunkers below the protective deck, pipes G (*see* plans) are led from them into the funnel casing; and to supply the partial vacuum thus created, fresh air flows through tubes H led into the bunkers from the ventilating shafts V. To supply fresh air to the spaces above and below the watertight platform W W, a steam fan S is stationed on the platform, and draws air from the foremost of the shafts V through a large supply trunk I. This air is forced by the fan into the trunk K and finds its way out through louvres into the compartments through which K passes. A number of pipes also lead from this trunk to supply the magazines, shell rooms, &c., as indicated in the Plate. Arrangements are made so that the trunk I can be connected to one of the steam fans in the foremost boiler room, which fan can therefore be used to ventilate the fore part of the vessel in the event of the one on the platform breaking down. The shell hoist P, primarily fitted for the conveyance of



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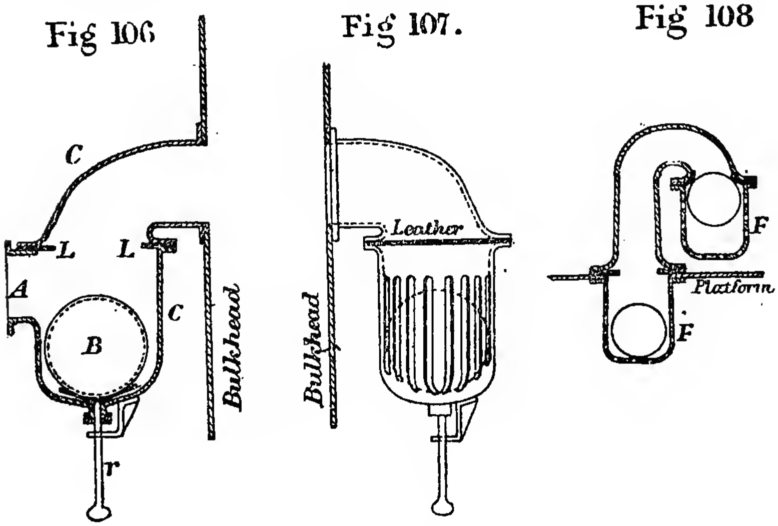
shells to the guns, is provided with a cowl, and thus conducts fresh air into compartments below the protective deck. Vitiated air in the compartments below the watertight platform is, in this vessel, exhausted into those above it either through the hatchways L or the small pipes N; and the spaces above the platform exhaust from one to the other in turn through holes cut high up in the bulkheads until the air finally reaches the under-water torpedo room, whence it is drawn into the boilers through holes cut at O in the fore bulkhead of the forward boiler room. It may be mentioned that the air pipes *a*, through which air escapes when flooding the wing and water ballast compartments, also assist in the ventilation of those compartments.

Self-acting or *automatic* valves are fitted where watertight bulkheads and platforms are pierced for ventilation purposes, in order to prevent water passing from a damaged compartment into adjacent ones. For example, if the compartment on the after side of the bulkhead *b b* became bilged, water might enter the trunk K through louvres and then flow into spaces on the fore side of the bulkhead if no precautionary measures were adopted. Where a ventilation trunk passes through a watertight bulkhead, an *automatic float valve* is provided on each side in the manner shown (for one side only) in section at Fig. 106. A casting C,

Fig 106 Fig 107. Fig 108



made in two parts, is fixed to each side of the bulkhead, the lower part of each being joined to the trunk at A and containing a hollow copper ball B, which is so disposed that air can pass freely over it from one side of the bulkhead to the other. If, however, water obtained access to the trunk, it would flow along until it reached one of the copper balls, and this would then rise and become pressed against the leather marked L,—inserted at the joint of each casting—thus securing watertightness. Communication between the compartments can also be cut off in the event of fire by pushing up the particular rod *r* which is farthest from the fire until the ball bears tightly against the leather, when the rod can be secured in position by means of a small pin. Again, when one compartment exhausts into another through a hole cut in the bulkhead (as at O, Plate XV.), float valves are adopted as illustrated in elevation by Fig. 107, the lower part of the casting having slots cut in it to admit of the passage of air; one of these valves is placed on each side of the bulkhead. Fig. 108 shows a valve fitted where a compartment below a watertight platform is exhausted into one above it. The parts F containing the floats are perforated, and the upper ball is shown in the position it would occupy if the space above the platform became flooded. In the event of compartments below that platform being filled, the water would be prevented from rising higher by the lower valve. A different arrangement is adopted for the exhausts from magazines; there are here no ball valves fitted, and, as will be seen from Plate XV., the foul air from each magazine passes into the compartment above through a pipe having its upper end turned over and perforated, whilst the end in connection with the magazine can be closed when necessary with a strong brass horizontal sliding shutter, in order to effectually resist fire from explosion on the outside. The plan on Plate XV. shows that float valves are fitted in the coal bunkers to the ends of the pipes G and H, each being surrounded by a shield to prevent damage. On the other side of the coal bunker bulkhead a slide valve T, worked by hand, is provided at each pipe to shut off communication in case of fire in the bunkers, or to prevent the escape of air, through the coal bunkers, when the boilers are being worked under forced draught; they would also be used in place of the automatic valves if the latter failed to act. It should be mentioned that similar valves are also fitted to the ends of pipes leading to and from the *upper* bunkers.

A brief reference will now be made to the methods employed in ventilating a recent composite sloop, where natural ventilation is almost exclusively adopted. Each store and provision room is provided with fresh air in the following manner. A cowl is fixed above the upper deck to lead into a space included between two frames, the bottom planking, and the thin steel lining which is secured to the inside of the frames. The air passing down this trunk is admitted to the store room through a sliding shutter, this latter having an air- and water-tight sill fitted beneath it between the frames, to prevent the ingress through the louvre of foul air from the bilges. Each of the above rooms is *exhausted* through a sliding shutter fitted between two other frames, the air passing out at a similar shutter affixed in the same frame space above the upper deck. Sometimes a separate stout pipe leads from the supply cowl to the compartment, and a similar pipe also used for the exhaust.

The magazines are supplied with fresh air by small ventilating fans worked by hand, from which pipes are conducted to open in the floors of the magazines. Foul air is got rid of by leading up between the frames a pipe from the crown of each magazine, gauze diaphragms being fitted in it—as usual for the magazines of all ships—to prevent any possible entry of combustible material into the magazine. These pipes either exhaust through louvres in the thin steel lining above the upper deck, or have their ends continued above the topsides and then arched over as an additional precaution against the entry of sparks, water, &c. In the latter case they, in common with all ventilation pipes leading from above the topsides to magazines, have about four feet of their length made of teak or vulcanite in order to break the continuity of the metal pipe; this is to avoid damage to the magazine should that pipe be struck by lightning. The part of the iron pipe above the break is put into metallic connection with the hull of the vessel so that the electric fluid can pass harmlessly into the sea.

The remaining ventilation of this vessel is the same in principle as the natural ventilation of the one previously described.

It need scarcely be remarked in conclusion that although the foregoing exhibits the main principles upon which the ventilating arrangements of a vessel are based, there are numerous differences of detail in different ships.

CHAPTER XIII.

STEERING ARRANGEMENTS.

THE following remarks on the above subject will be confined to stern rudders and the gear for actuating them, omitting all reference to the auxiliary appliances (such as bow, and additional stern rudders) which have occasionally been fitted with a view to increasing the steering qualities of ships.

Plate XVI. shows at R the two kinds of rudder now employed in the Royal Navy. That given in Fig. 109 is the *ordinary* rudder, hung by pintles at its forward or leading edge to the lugs attached to the rudder post. Such a rudder has the whole of its area abaft the axis about which it rotates, and therefore the point where the resultant water pressure upon it acts, when the vessel is in motion and the rudder turned out of the middle line plane of the ship, is well abaft that axis. The pressure has thus a good leverage about the axis, and considerable effort is necessary to hold the rudder over to large angles. With the *balanced* rudder, partly shown in Fig. 110, the case is different. Here about three-eighths the total rudder area is placed on the fore side of the axis; and since the forward part of a rudder is the more efficient, the point of application of the resultant water pressure on the total area lies very close to the axis of rotation, and the rudder can be held at large angles with comparatively slight effort. Herein lies the superiority of the balanced, as compared with the ordinary rudder, since large areas can be employed, and good turning power thus ensured, without necessitating a very large force to maintain it in position. They were introduced into the large vessels of the Royal Navy many years ago, in consequence of the long time consumed in getting over to extreme angles the ordinary rudders of such ships by the hand gear then employed; and they successfully fulfilled their object. It was, however, found desirable to use a smaller rudder area when proceeding under sail alone, and this led to the introduction of the "compound" balanced rudder, in which the area before the axis could be moved independently of the after part, and could be locked amidships when the vessel was proceeding under sail, the after part being then used as an ordinary rudder. With the advent of *steam* steering gear the reason for using the balanced rudder disappeared, and the ordinary type was

Fig. 109.

SIDE ELEVATION.

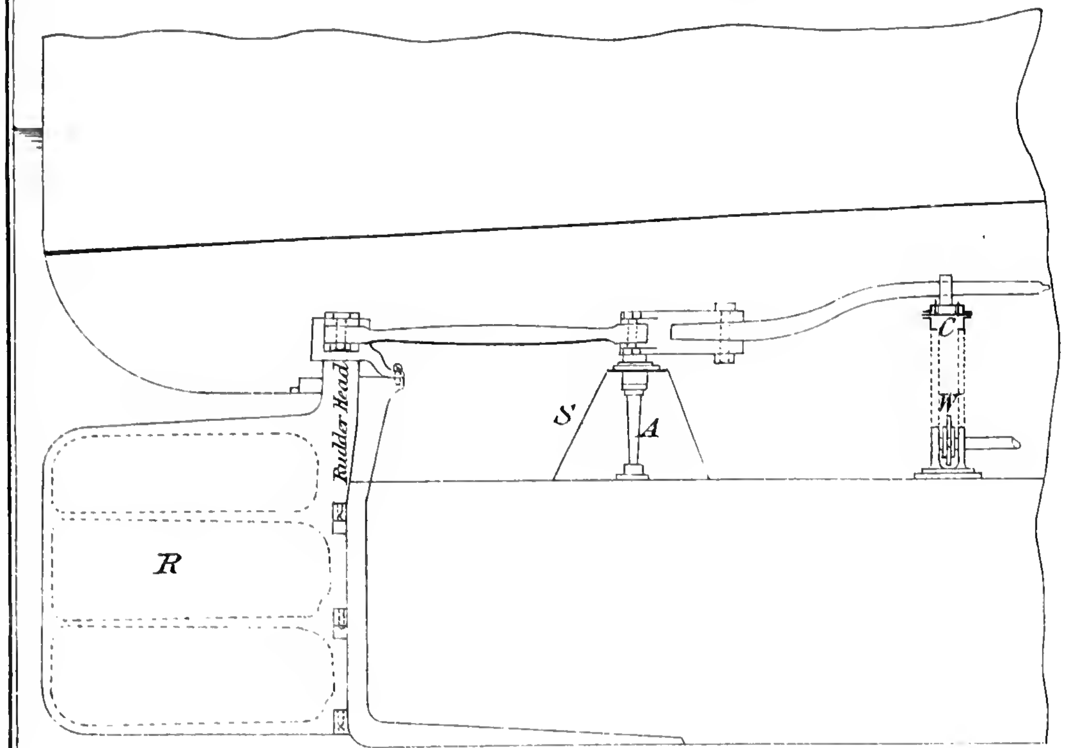
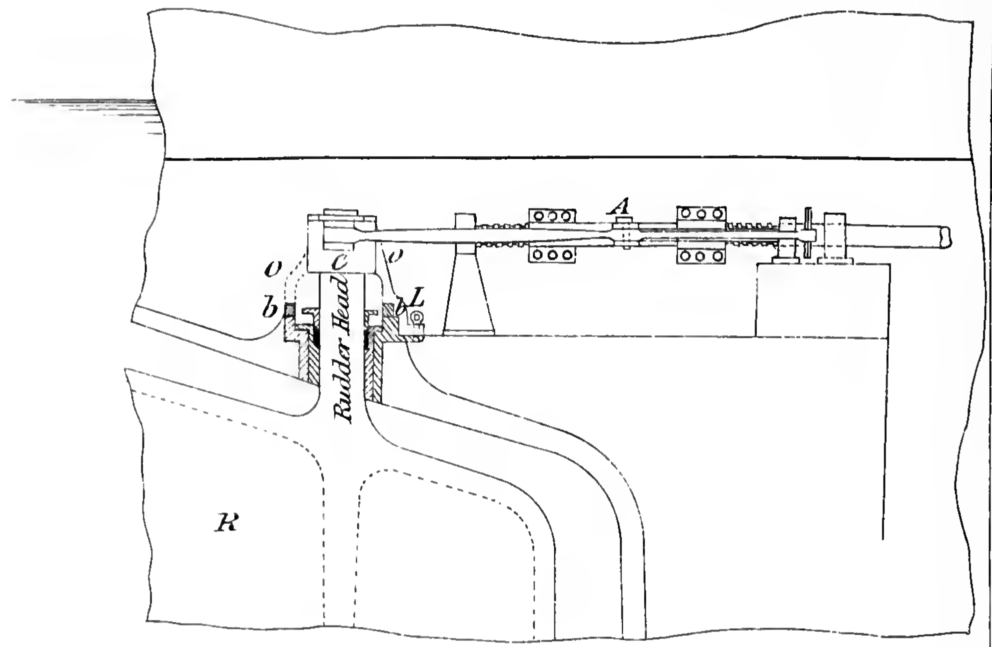
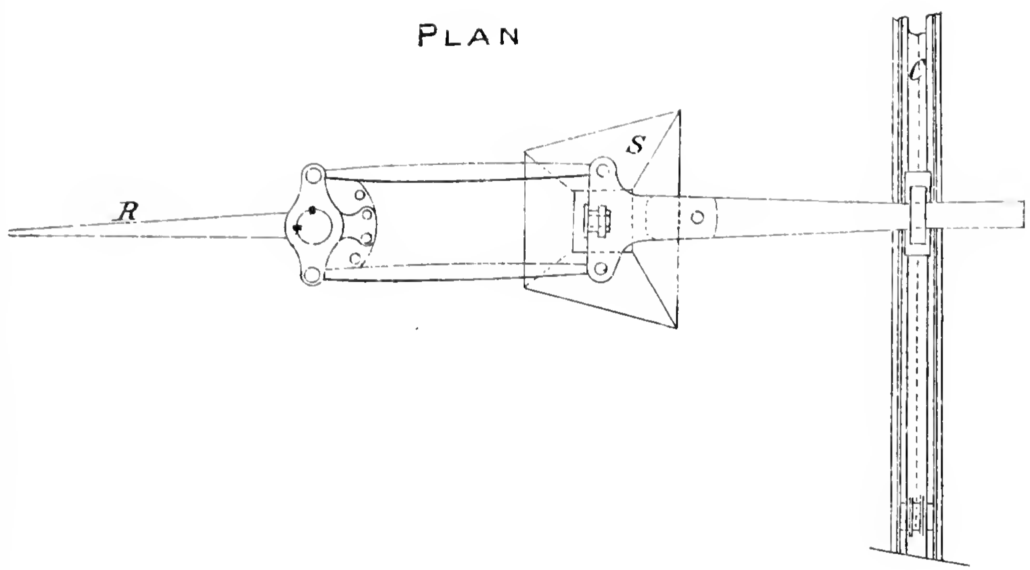


Fig. 110.

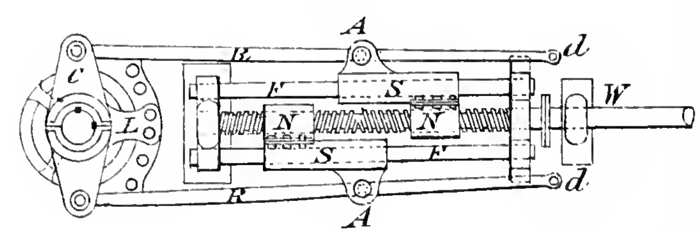
SIDE ELEVATION



PLAN



PLAN.



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reverted to. Very high speeds have, however, been given to recent vessels, and as increase of speed greatly augments the pressure on the rudder, balanced rudders have in some cases been re-introduced to avoid the very strong and heavy appliances which would otherwise have been necessary for holding the rudder in position at extreme angles.

Fig. 109 shows the general method of framing the rudder of an iron or steel ship. It is usually of wrought iron, and the head and front part of the frame are made in one piece, in which the pintles for carrying the rudder are formed. There are three of these pintles in the rudder shown, of which only the middle one passes completely through the corresponding lug on the sternpost. Into the ends of each of the others a small hemispherical point of hardened steel is screwed, and these bear upon similar points in the stern post, thus taking the weight of the rudder without producing much resistance to its rotation. The framing of the back part is joined to the front portion, and consists of an outer rim bent to the shape of the rudder, with two supports between it and the front edge, as shown by ticked lines. The rudder framing tapers in thickness from the front edge, as in the plan, and wrought iron plates are tap riveted to it in order to form the sides of the rudder, the space between them being filled up with some light wood, generally fir, to prevent the entry of water. For balanced rudders similar methods of construction to the foregoing obtain, it being observed that the only pintle used in a balanced rudder is placed on its lower edge, opposite the head. Cast steel is now often adopted for the rudders of iron and steel ships, and then the frame is often made in one piece. In vessels sheathed with wood, and coppered, the above materials could not be employed for these parts, for reasons already stated in Chapter VII.; in the largest sheathed ships, therefore, the rudders are made entirely of gunmetal (or, in the latest vessels, of phosphor bronze), whereas the rudder of a corvette or sloop is generally of wood, with a gunmetal or phosphor bronze casting at its upper part to take the rudder head, although rudders made entirely of phosphor bronze are now being used in these smaller vessels also.

A stuffing box is fitted where the rudder head passes through the stern of the vessel in order to secure watertightness; one of these is shown in detail in Fig. 110. From the same figure it will be seen that the weight of a balanced rudder is taken inboard by the steel casting O, which is rigidly attached to the rudder head and which bears upon, and slides in contact with, the sternpost at *b b*. This casting, and therefore the rudder, can be locked in various positions, if required, by the locking pins L which pass through it and into the sternpost. Instead of the sliding contact just mentioned, the weight of the rudder is sometimes taken upon several small rollers to diminish the friction as much as possible. The lower pintle of the balanced rudder is simply for steadying purposes, it being received into a spur projecting from the lower part of the sternpost.

The size, shape, and general arrangement of the gear for actuating the rudder differ greatly according to circumstances. Fig. 111 shows these details (excluding the steam steering engine and hand gear) in plan for a small ship. The rudder head A is clasped by the tiller, and the latter is prevented from slipping round the former by cutting vertical grooves out of each in corresponding positions and fitting *keys* K therein. Any movement of the tiller must thus affect the rudder, the latter being turned to starboard when the former is moved to port, and *vice versâ*. The forward end of the tiller is made rectangular in section and of constant size for about 5 feet of its length, and passes through a *Rapson's slide*, *i.e.*, a metal block provided with a vertical spindle which revolves in the carriage C. To move the tiller from side to side, C is joined to the steam and hand steering gear by steel wire ropes which lead round the system of pulleys shown. At their forward ends these ropes are joined by a chain which passes round a horizontal wheel, the circumference of which is formed to take hold of the links of chain. By turning this wheel therefore by the steam steering engine, or the hand wheels, the tiller is moved to starboard or port. As the tiller approaches the side, the carriage C slides forward along it, and thus causes the force pulling the tiller over to act at a greater leverage from the rudder head than would otherwise have been the case.

When a vessel is fitted with a lifting screw, the necessary aperture renders the above arrangement inapplicable. Fig. 112 shows, in plan, a method sometimes adopted in such a case. The *cross-head* D is keyed to the rudder-head, and a second cross-head E, connected to the tiller, is secured to a pivot or *auxiliary rudder-head* C, the two cross-heads being joined by the parallel rods R. These rods thus communicate any movement of the tiller to the rudder to precisely the same extent as would have been the case had it been possible to connect the tiller directly to the rudder-head.

Again, when a ship is provided with a complete protective deck, which is well below water at the ends, the steering gear is placed below that deck (where the vessel is necessarily narrow) for protective purposes. When this is the case, the width of the ship

Fig. 111

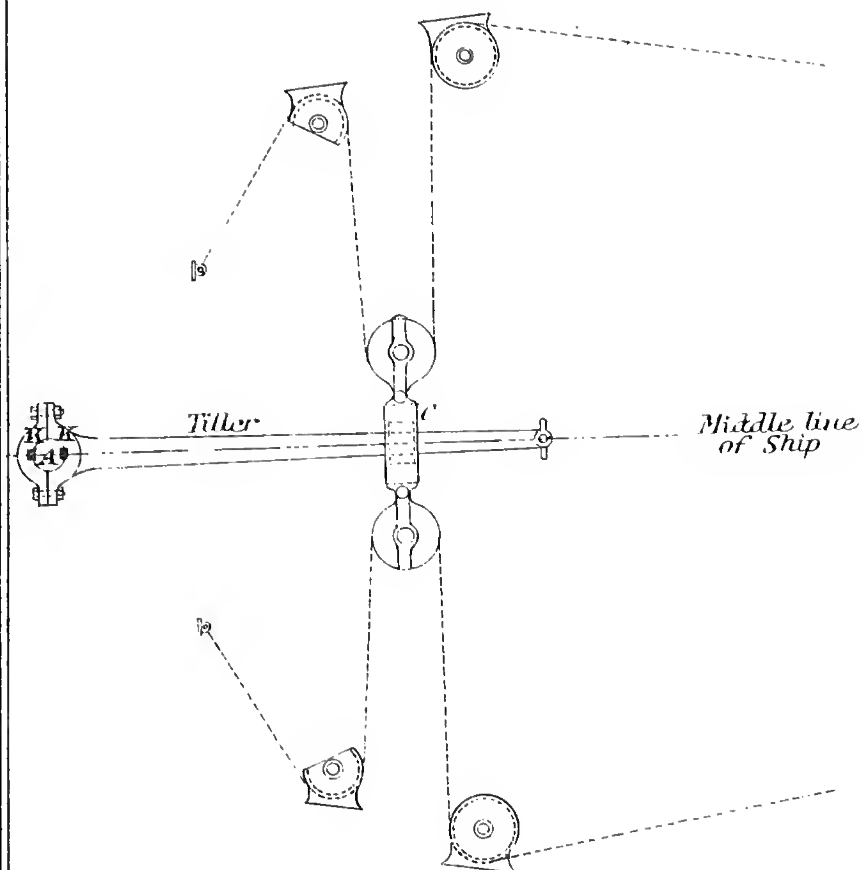


Fig. 112

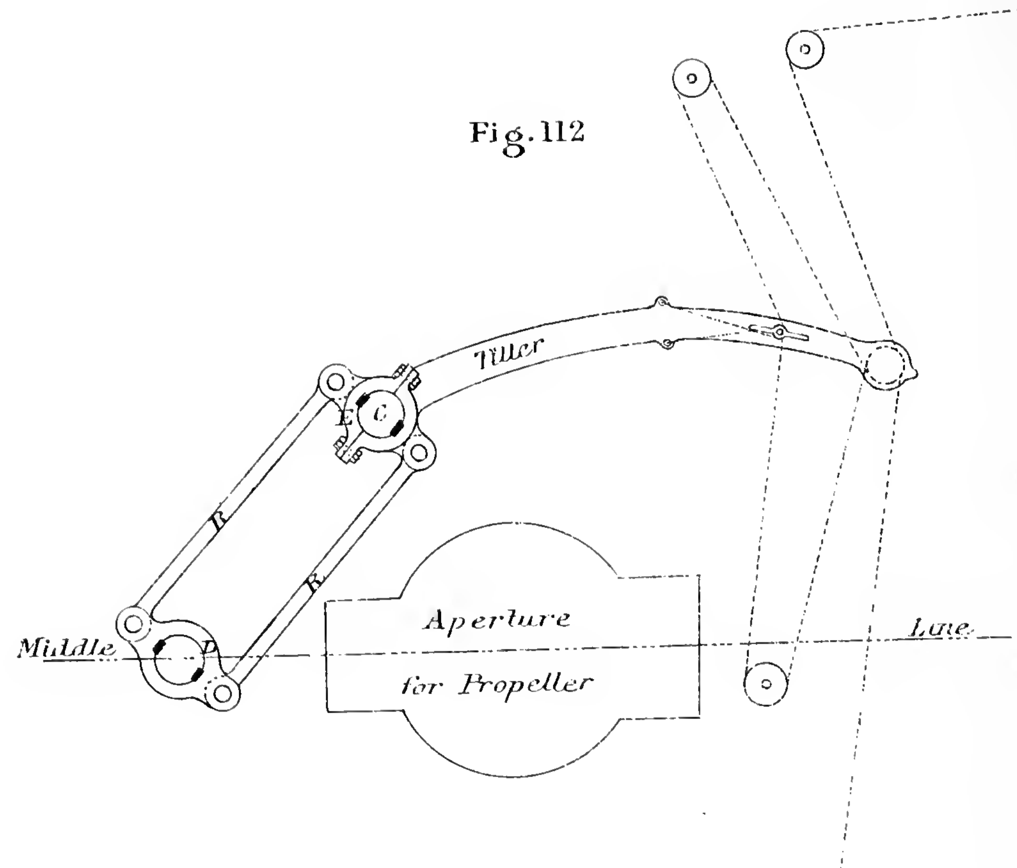
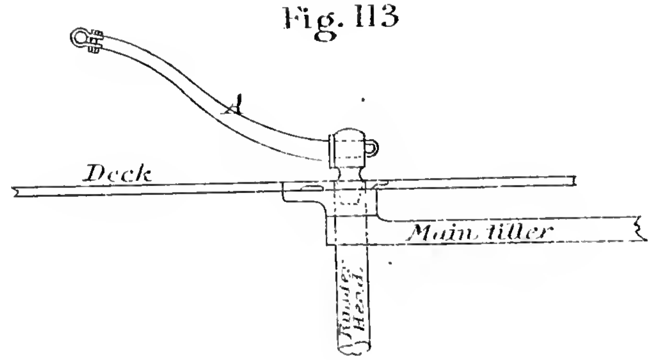


Fig. 113



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is usually insufficient to admit of an ordinary tiller being put over to the extreme angle desired, and then the parallel bar arrangement shown in Fig. 109 is generally adopted, or the screw gear sketched in Fig. 110. In the former an auxiliary rudder-head A is fitted, supported by the structure S; and the fore end of the tiller passes through a carriage and block similar to that in Fig. 111. In this instance, however, the carriage is provided with wheels, and runs on a carriage-way C, worked transversely across the ship. Each end of the carriage is joined to a chain passing round a sheave at each end of C, and thence round the sprocket wheel W, which is worked by the hand and steam steering gear.

The details of the apparatus shown in Fig. 110 are as follow :— A cross-head C (which is made in one with the part O, already mentioned) is well keyed to the rudder and attached to the connecting rods R, each rod being joined at A by a single bolt to the sleeve S, through which a fixed guide rod F passes. Each sleeve is free to slide along the corresponding guide rod, and is firmly bolted to a nut N, in which works a screw attached to the shaft W. It will be noticed that one half the length of the screw is right handed and the other half left handed, so that as W is revolved by hand or steam the nuts simultaneously approach towards or recede from each other, thus turning C, and therefore the rudder. The advantage attending the use of this gear is that, since a screw is practically *non-reversible*—that is to say, no likely pressure on the nut in the direction of the axis of the screw will cause the screw to turn—the rudder will remain in the position in which it is placed by the screw, even though it receives severe blows from the sea. No extra assistance is therefore needed at the wheels in heavy weather when steering by hand, and all possibility of the wheels flying back and so injuring the men is avoided. Also, when using the steam steering engine the effect of a blow on the rudder would only be felt by the screw and would not be transmitted to the engine.

Auxiliary means are always provided for moving the rudder of a vessel in the event of accident to the main apparatus. For example, if the steering engine and ordinary hand gear of a ship usually steered by a tiller broke down, a *relieving tackle* would be employed on each side. One block of each tackle is secured to a shackle on the end of the tiller, and the other block is fastened to an eye plate at the side of the ship; the tiller can then be moved to and fro by hand. Sometimes an *auxiliary* tiller is fitted, as indicated in elevation at A in Fig. 113; this can be shipped into the rudder head and worked if the main tiller breaks down from any cause. When *screw* steering gear is employed, a tiller is also provided for use in case of necessity. It will be seen in the plan of Fig. 110 that the rods R are prolonged to *d* in order that, if anything happened to the screw gear, these ends may be attached to a tiller (working round an auxiliary rudder head, not shown in the figure) after the rods R have been detached from the sleeves, thus producing an ordinary parallel bar arrangement. This tiller is fixed just above the screw gear. Sometimes provision is made for joining the ends *d* together after the bolts at A have been removed, the rods being then used as an ordinary tiller.

In recent ships telegraphic communication has been electrically arranged between the various working or conning positions, the steering positions, and the rudder head. An electrical *transmitter* is placed at each working position, by means of which the commanding officer is enabled to transmit to any particular wheel elsewhere (say to an underwater protected steering position) the angle of helm which he desires. This order is received on a dial or *receiver* at the steering position, and the wheel is put over to the side and degree required. In order that the officer and the man at the wheel may both be sure that the desired angle of rudder has been obtained, an *automatic* electrical reply is arranged at the rudder head which indicates on dials or *indicators* at the conning and steering positions the actual angle of helm given. The steersman moves his wheel in the required direction until the actual angle recorded on his "indicator" from the rudder head corresponds with the angle of the order on his "receiver" from the officer; at the same time the officer's "indicator" shows him the same angle, and satisfies him that the order is carried out.

Switches are arranged at the various instruments, so that only those required to be in use are working at the same time.

CHAPTER XIV.

PROTECTION OF SHIPS AGAINST GUN ATTACK.

In this chapter it will only be necessary to notice the character of the protection given to all war vessels—in a greater or less degree —in order that they may resist gun attack, the measures taken to ensure their safety when assailed by a ram or torpedo having been already pointed out in connection with the watertight subdivision of ships.

It is needful to shield against excessive damage from this cause the parts securing buoyancy and stability to the ship ; and at the same time to preserve intact such vital parts as the machinery, magazines, and steering gear, as well as the big guns of large vessels. Buoyancy must be protected by preventing shot and shell from damaging the skin of that part of the vessel which is situated below water, whether the missile first strikes above the water line or, when the vessel is rolling, below that line for the upright position ; and the arrangements adopted for this purpose in the neighbourhood of machinery, magazines and steering gear, equally protect these parts also. To maintain sufficient stability, it is important to preserve uninjured a large proportion of the water-plane area and reserve of buoyancy, since any diminution of the former would lessen the metacentric height, and, therefore, the power of the vessel to stand upright ; and any reduction of the latter would lead to decreased stability at large angles of inclination.

Protection is attained by employing thick iron or steel plates on the side of the vessel or in the form of protective decks, by utilizing the coal supply, or by a cellular subdivision of the vessel, these compartments being sometimes packed with some light material to exclude water should the divisions become pierced, and at others being empty or utilized for the stowage of stores. These systems will now be referred to in detail.

Ships provided with thick side armour are known as *armoured* vessels ; several of these are depicted in Plate XVIII., from which can be gathered the amount of protection afforded in each ship to the several parts enumerated above. The first English armourclads were the small vessels *Thunderbolt*, *Erebus*, and *Terror*, built during the progress of the Crimean war ; the thickness of armour employed was about $4\frac{1}{2}''$. The larger vessels, *Warrior* and *Black Prince* (*see* Fig. 114), followed, and had a partial belt of armour $4\frac{1}{2}''$ thick, this being capable of resisting projectiles from the heaviest guns then afloat. These vessels were open to the objections that their ends and steering gear were unprotected, and hence the *Minotaur* class were made large enough (10,700 tons) to carry a complete belt of armour from stem to stern ; this is $5\frac{1}{2}''$ thick amidships, but thinner towards the ends. About this period more powerful guns were supplied to vessels, and hence a corresponding thickening of the armour to resist them became necessary. To accomplish this, without greatly adding to the weight of armour, the "belt and battery" system was adopted. This consisted in having a complete belt at the water-line, and a central battery amidships, into which the fewer but heavier guns could be concentrated. This shortening up of the space requiring to be armoured in order to protect the guns, made it possible to increase the *thickness* of the armour used. The *Bellerophon*, Fig. 115, exemplifies this distribution of armour and armament, the former being $6''$ thick. The *Penelope* and *Hercules* are of the same type, the latter being provided with armour $9''$ thick amidships, and $6''$ thick at the ends. Succeeding ships had guns placed in an upper deck battery immediately over the lower one, a good gun fire right forward and aft being thereby secured ; this is typified in the *Alexandra*, Fig. 116, which has $12''$ armour on the water-line belt—tapering to $10''$ at the ends—and $8''$ and $6''$ plating on the batteries. A $1\frac{1}{2}''$ iron deck joins the top of the comparatively low belt armour on each side, to give protection against high angle or depressed gun fire. The *Téméraire* followed, this vessel having her upper deck armament placed on turn-tables in two fixed armoured redoubts or *barbettes*, one near each end of the ship ; her water-line protection is similar to that of the *Alexandra*. In the cruiser *Shannon*, Fig. 117, the guns are unprotected by armour, except by a bulkhead $8''$ thick, worked across the gun-deck forward to screen them against a raking fire ; and the $9''$ armour at the water-line is stopped 60 feet short of the stem, its ends at this point being joined across the ship by an armoured bulkhead. Forward of this, a $3''$ iron under-water protective deck was fitted, which was the first example of a now very general method of protection ; a deck, $1\frac{1}{2}''$ thick, also joined the top of the belt armour. The later cruisers, *Nelson* and *Northampton*, are similar to the preceding, except that an under-water deck (3″ thick) and armoured screen bulkhead were given to *each* end of the ship; and the deck above the armour was 2″ thick. The growing size and power of naval guns caused further concentration of the armament of large ships, and the central battery gave way to the plan of placing the guns in revolving towers or *turrets*, by which means the heaviest guns can be easily trained through large arcs, and the same guns used on either side of the ship. Fig. 118 represents the *Dreadnought*, which has a complete belt of armour from stem to stern, and a *central citadel* rising above this in which the turrets are placed; the maximum thickness of armour is 14″. All recent first-class battle ships have their principal armament concentrated into turrets or barbettes, as exemplified by Figs. 119, 120, and 51 respectively; but to resist the powerful guns now carried by vessels, it is necessary to provide thicker armour than was given to the *Dreadnought.* To do this without undue increase in the size of ship, the ends at the water-line are left unprotected by vertical armour, and the weight so saved is utilised to thicken the remaining side plating; strongly plated under-water decks at the ends preserve from damage the spaces below them. Thus, the *Inflexible*, with a short water-line belt of the same length as the central citadel, has a maximum thickness of armour of 24″ at the water-line, and 17″ on the turrets; the under-water deck at ends as also the deck over the citadel, being of iron 3″ thick. The parts immediately underneath turrets must be protected by armour in order to shield the turret-turning gear; but this is not so necessary beneath barbettes (except as regards the ammunition trunk) since the barbette itself covers the machinery for actuating the turn-tables. In barbette ships, therefore, a longer though narrower belt is possible on the water-line without increasing the weight of armour as compared with a turret ship; this will be seen from Fig. 51, particulars of that vessel having been given in Chapter V. This allows of the guns being placed farther apart, and, being in separate protected stations, they are not liable to be all disabled by the explosion of a single shell, as is possible when they are placed in the same contracted central citadel, as in the *Inflexible.* The guns can also be carried higher out of the water than is possible in a central citadel turret ship, on account of the extra weight of armour which would be necessary to protect the turret-turning gear. In the *Trafalgar* the maximum thickness of armour is 20″, and the greater displacement of that vessel enables a longer citadel and still longer belt, as compared with the *Inflexible*, to be given to it without undue sacrifice of other qualities. The secondary armament of this vessel, placed above the citadel, is protected against a raking fire by a 5″ armoured bulkhead at each end of the battery, and against fire on the broadside by 3″ armour, in addition to the 1″ skin plating. The *Sans Pareil* and *Victoria* have a distribution of armour differing somewhat from the preceding. They are provided with one turret only, and have a partial armour belt similar to, but longer than, that of the *Admiral* class ; from the 3″ armoured deck over the belt, at its forward end, a barbette or redoubt rises, and encloses the base of the turret, thus protecting the turning and loading gear.

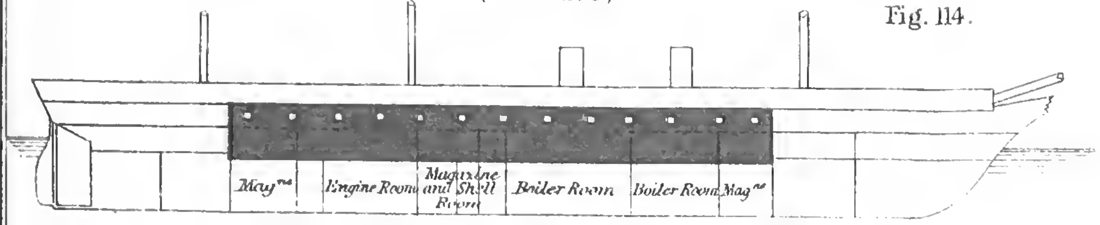
Unarmoured vessels having a protective deck—as the *Mersey*, Fig. 121—are known as *protected* ships ; in the later vessels this deck extends from stem to stern, but in earlier ships it covers the engines and boilers only, as in the " C " (Fig. 52), and other classes. The earlier ships also had their deck entirely below the level of the water, whereas later ones have theirs at or above that level at the middle line of the ship for a good proportion of its length. The decks of such vessels are necessarily below water at the sides, as shown in midship section of *Mersey*, in order to prevent projectiles striking between wind and water from entering into the compartments below. Fig. 121 gives the thicknesses of the *Mersey's* deck, and the maximum thickness of the large protected cruisers *Blake* and *Blenheim* is 6″.

In all vessels the armour was made of one thickness of plate until the time of the *Inflexible*. In that vessel the side and turret armour was made up of two thicknesses, separated from each other by a layer of wood ; this constitutes the *sandwich* system of armour-plating, and was adopted for the side armour of the similar, though smaller, vessels, *Ajax* and *Agamemnon*. The turret armour of the last-named vessels, as well as all the vertical armour of later ships, is in one thickness only.

The material employed for the armour of all ships down to the *Inflexible* and the similar vessels mentioned above, was *wrought iron ;* but on the turrets of those vessels, *compound* or steel-faced plates were introduced, and this compound armour has, since that time, been exclusively adopted for all thick armour plating. It consists of a *back* or *foundation* plate of wrought iron, faced by a layer of hard steel (*see* Figs. 123 and 124), the two being fused

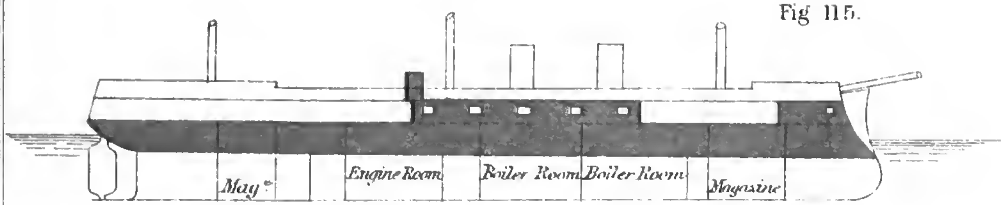
WARRIOR
(9200 tons)

Fig. 114.



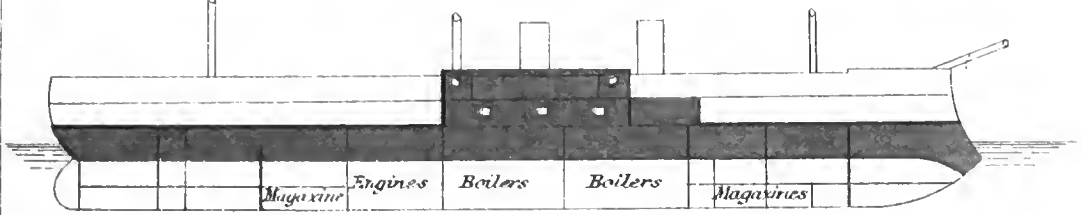
"BELLEROPHON"
(7550 tons)

Fig. 115.



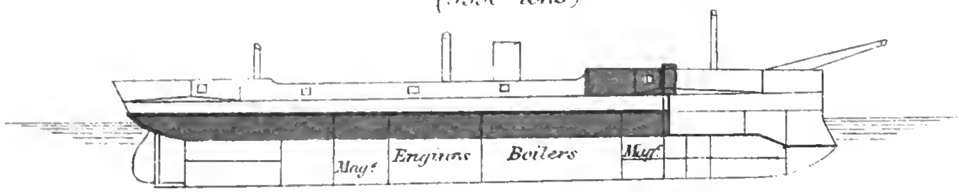
"ALEXANDRA"
(9500 tons)

Fig. 116.



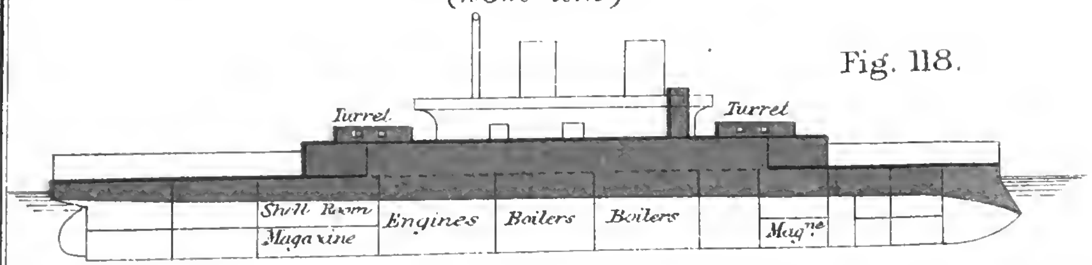
"SHANNON"
(5390 tons)

Fig. 117



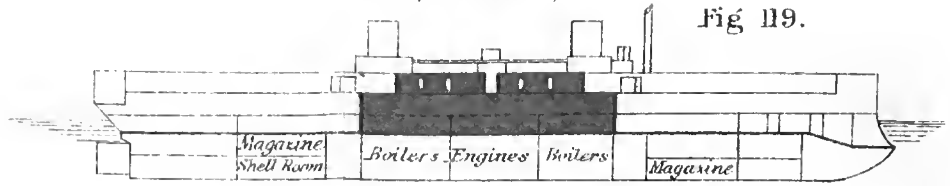
"DREADNOUGHT"
(10820 tons)

Fig. 118.



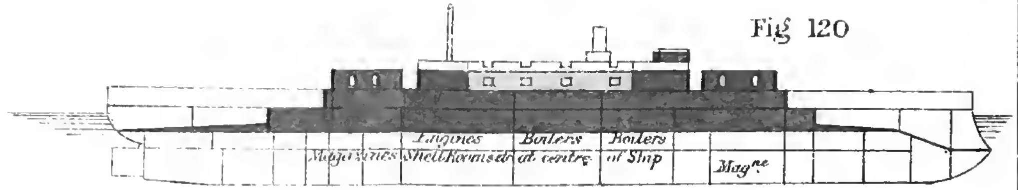
"INFLEXIBLE"
(11880 tons)

Fig. 119.



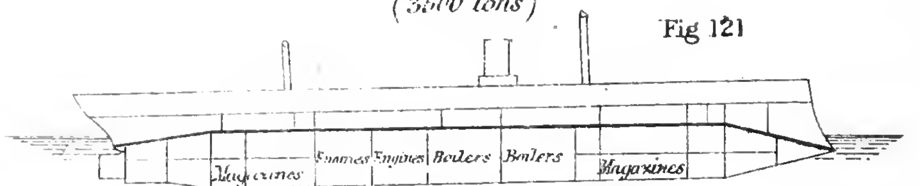
"TRAFALGAR"
(12500 tons)

Fig. 120

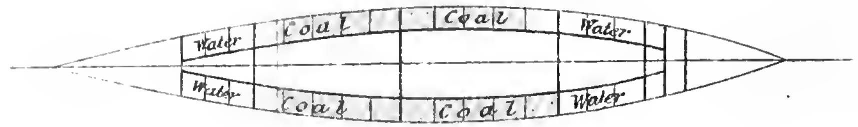


"MERSEY."
(3500 tons)

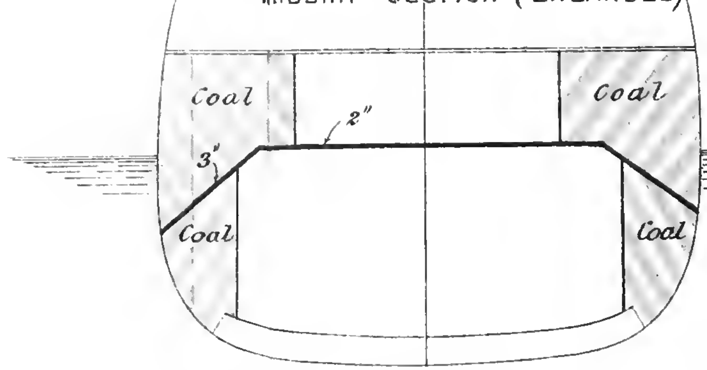
Fig 121



"MERSEY"
PLAN OF PROTECTIVE DECK.



"MERSEY."
MIDSHIP SECTION (ENLARGED)



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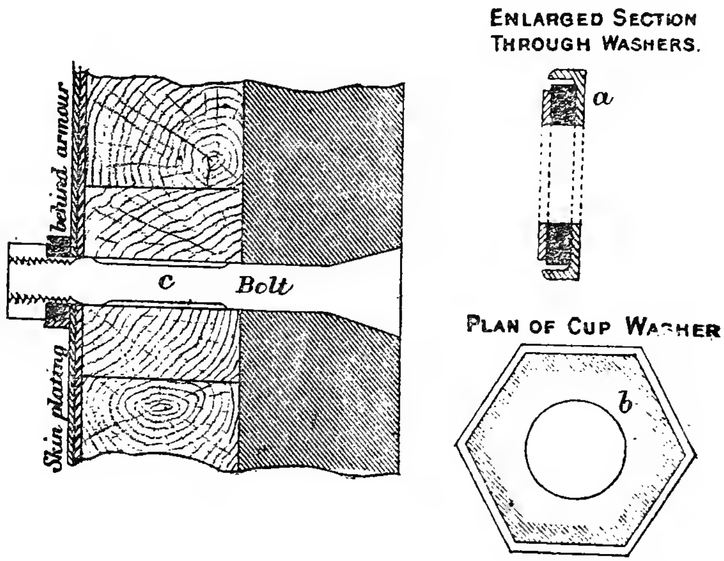
together. The thickness of the steel is usually one-third the total thickness of the plate ; and the reasons leading to the adoption of this armour were as follows :—A hard plate was required in order to break up the shot on impact, and thus attention was early directed to steel ; plates composed wholly of that material were, however, found to crack very much, although the shot was broken up. On the other hand, plates of ductile wrought iron did not crack, but allowed the projectile to get through intact, and these facts led to the trial of compound armour, whereby the requisite hard surface was obtained in combination with ductile wrought iron, the latter preventing excessive cracking. These compound plates are at least 20 per cent. stronger to resist the attack of shot than the same thickness of wrought iron. It should be mentioned, however, that in recent years great progress has been made in the manufacture of armour plates composed *entirely* of steel, and these have been adopted in some vessels of the French and other navies, good results having been obtained on trial. Protective decks, which were at first of iron, are now constructed of *mild steel.*

Between the skin-plating and the armour, a layer of *wood backing* is placed, composed of teak; this is shown in Figs. 63 and 73. It is well secured to the skin plating by flat-headed bolts, which are either screwed into that plating or have nuts on their points ; and its functions are to distribute any blow given to the armour over a large area, so that local injury to the framing may be lessened ; also to act as a cushion, thus deadening vibrations caused by a blow, which vibrations would injure the armour fastenings ; and, in conjunction with the skin plating, to prevent splinters of plate, shell, &c., from entering the ship. Its thickness is 15″ in the *Admiral class*, but only 6″ behind the 18″ armour of the *Victoria*, and 4″ in rear of the 20″ armour of the *Trafalgar*. These two last-named vessels have the strong framing behind armour sketched in Fig. 63, and the wood backing is made as thin as possible in order to diminish the elasticity of the armour support, thus helping to procure that rigidity which has been found necessary for the supporting structure of compound plates in order to bring out their maximum powers of endurance ; the weight thus saved from the backing is utilised for the strong plate frames.

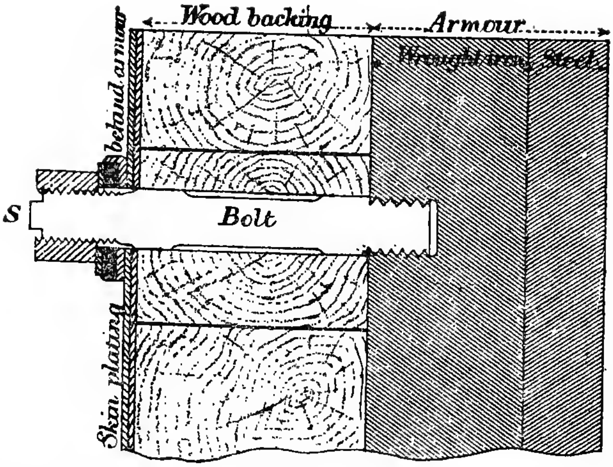
Wrought iron plates were fastened to the ship by wrought iron conical-headed bolts, which passed through the armour, and had nuts on their points hove up against the skin plating behind armour on *elastic cup washers*, as shown in Fig. 122. Each of

Fig 122



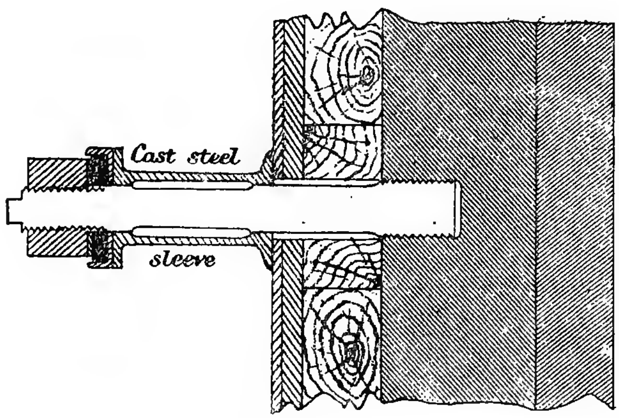
these washers consists of a hexagonal cup washer, marked *a* and *b* in the enlarged section and plan respectively, in which fits loosely a similarly-shaped indiarubber washer, covered by a thin plate washer ; it is against this latter that the nut of the bolt presses The elastic washer was introduced in consequence of the liability of the nuts to fly off under the jarring effect of a blow upon the armour, and has greatly lessened that liability. For a somewhat similar reason the shank is reduced at *c* to a diameter slightly less than that at the base of the screw thread. Before this was done, bolts almost invariably broke off at the thread when subjected to severe jarring stresses, since they were weakest at that part ; but by introducing a long slightly weaker part *c*, all the stress is taken by it, and the bolts only *stretch*. A bolt for compound armour is sketched in Fig. 123. It does not pass right through the plate, because any holes in the steel surface would render the plate very liable to crack if struck by shot. These bolts are of mild steel, and are screwed into the back plate from inside the ship by means of a spanner, which fits over a square projection *S*, formed on the

Fig 123.



inner end of each bolt. As will be seen, an elastic washer is used as before. Where very thin backing is adopted, the small distance separating the armour and skin plating would preclude any reduction in diameter of the bolt shank, except for a very small length, if the ordinary method of fastening were carried out. To retain the advantage of the reduced shank, these bolts are hove

Fig 124.



up on a cast steel sleeve 12″ long, inside the skin plating, as shown in Fig. 124.

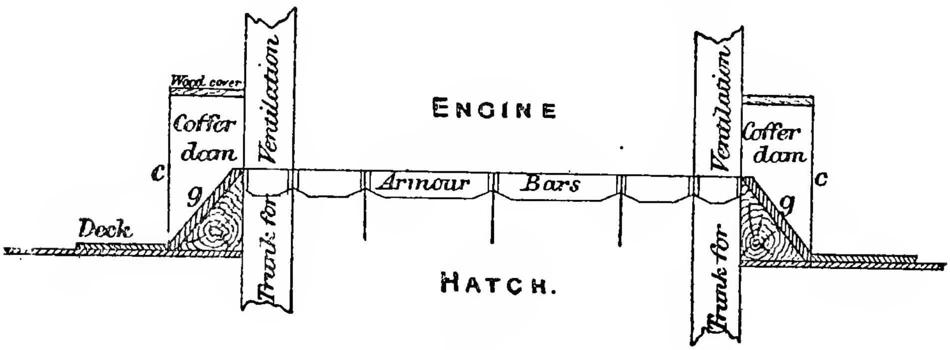
No edge or butt connection is made between any armour plate and adjacent ones.

Wrought iron plates had their fastenings disposed round their edges, and, to maintain these plates in position, as few bolts were used as possible. The bolts for compound armour are, however, more numerous, and are evenly distributed over the surface of the plate, so that even if the latter should break up under a very heavy blow into several pieces—as steel-faced plates are liable to do—the fragments would still be held in place by the bolts, and would thus be available to resist further attack.

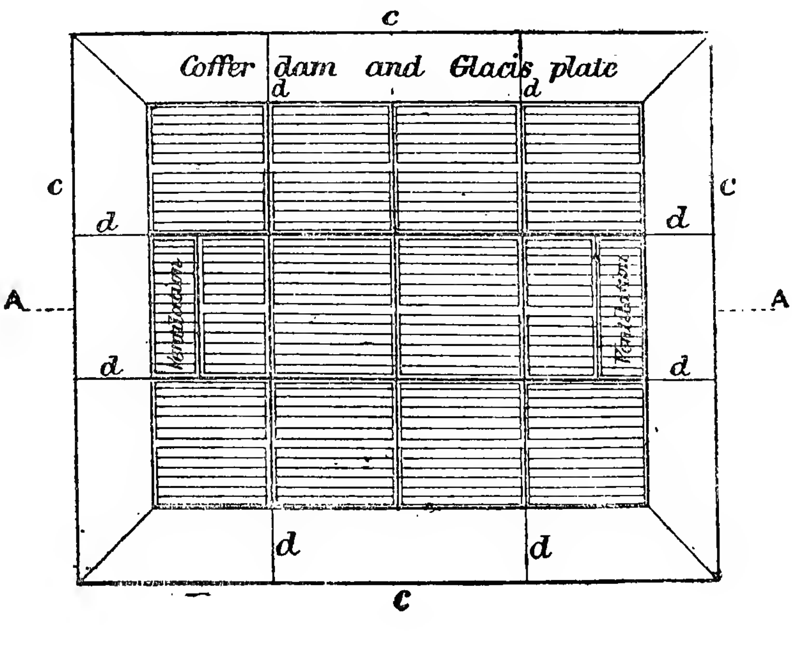
All openings in protective decks which cannot be closed in action, such as the funnel and ventilation hatches, are provided with armour bars or gratings, to prevent the entry of shot and shell. Engine room hatches are sometimes similarly fitted, a sketch of one of these being given in vertical section and plan at Fig. 125. The armour bars are about 8″ deep, $\frac{5}{8}$″ thick and $2\frac{1}{2}$″ apart in a

Fig 125.

VERTICAL SECTION THROUGH A.A.



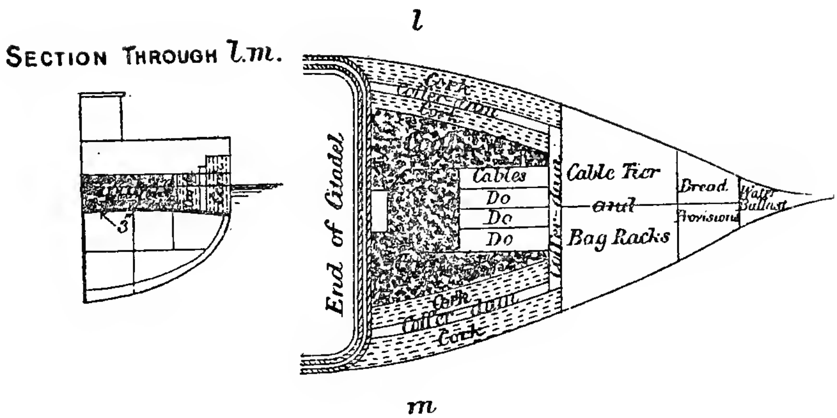
PLAN



large ship; and their ends either fit into grooves cut in supports fitted to receive them, or several bars are joined together to form a grating. These openings are generally further protected by working sloped armour plates—called *glacis plates*—around them about 3″ thick, as at *g* in the section. In addition to the above, where a deck is in the vicinity of the water-line, such openings have a *coffer dam* fitted round them, as in the figure. This is formed by placing a thin vertical plate *c* round the hatchway, and from 1 to 2 feet from it, its height depending upon the position of the protective deck relatively to the water-line. The space between *c* and the hatch is subdivided by vertical partitions *d*; and, in the probable event of water obtaining access above the deck during an action, it can be prevented from flowing down the hatchways through any shot holes made in the hatch casing in the vicinity of the coffer dam, by packing canvas, hammocks, &c., in the latter in the neighbourhood of the holes. Openings in protective decks which *can* be closed in action are provided with armour shutters and scuttles, made watertight by the use of india-rubber.

Protection by cellular sub-division, in order to localize the effect of any one shot, is adopted in conjunction with an armoured deck below, or in the vicinity of, the load water-line. For example, the space between the thick steel and lower decks of the "C" class (*see* Plates II. and IV.), is divided by the longitudinal girders and transverse partitions into 44 compartments, including the coal bunkers at side; and the bunkers above the protective deck of the *Mersey*, shown in Plan at Fig. 121, and other unarmoured vessels, as well as those fitted in many armour-clads above the protective deck amidships, are divided by transverse bulkheads into about 12 feet lengths for the same reason. Also, as already pointed out in Chapter X., similar sub-division is adopted above the under-water decks at the ends of central citadel ships. In the case of the *Ajax* and *Agamemnon*, some of the cellular divisions on the under-water decks were filled with cork, in order to lessen, in conjunction with stores such as coals, the space to which water could find access in the event of the unarmoured side being damaged near the water-line. Fig. 126 gives a plan of

Fig. 126.



protective deck at the fore end, with a section, the arrangement at each end being practically the same. Two belts of cork, each 4 feet wide, are worked at each side of the ship in watertight chambers for a length of 30 feet forward and 37 feet aft from the citadel, and are separated from each other by a coffer dam, 2 feet wide. Experiments demonstrated that a shot hole through cork does not close up after the passage of the shot, and hence the necessity for the coffer dam, into which canvas and oakum may be pressed in the neighbourhood of a hole. The *Inflexible* also had a somewhat similar arrangement of cork, but none of this has been fitted in later ships as they have a longer armour belt at the water-line and a smaller space accessible to water at the ends. In some French and other vessels, *cellulose* (a preparation of cocoa nut fibre) has been used instead of cork for filling up cellular divisions, this material, it is averred, having the property of closing up after a shot passes through it, thus preventing the entry of water. The results of experiments conducted with it in this country have not, however, been such as to secure its adoption in the British Navy.

To ascertain whether it was possible to utilize the coal supply of merchant and other ships for protective purposes, experiments were carried out at Portsmouth and Shoeburyness. These showed that loose coal, 20 feet thick, would stop the projectile from a 6″ gun at short ranges, and that 30 feet would resist the 8″ gun; also that, speaking roughly, 2 feet of coal were equal in resisting power to 1″ of iron. It was also found from these experiments that the coal was not set on fire by the explosion of shells in it, and that such explosion had little effect in displacing the coal. Examples of this coal protection will be seen in Figs. 76, 80, and 121.

CHAPTER XV.

THE PRESERVATION OF SHIPS.

In order to preserve the iron or steel hull of a vessel from rapidly wasting away through corrosion, galvanic action, etc., several precautions are necessary, the principal of which will now be briefly referred to.

As soon as possible after each piece of iron or steel work for the hull is completed, it is thoroughly scraped and cleaned to remove all traces of scale and rust, unless, for special parts, the "pickling" process described in Chapter IV. is resorted to; the iron or steel is then thinly coated with paint to prevent oxidation from exposure to the weather during the building of the ship. This removal of scale is necessary, not only for the reason given in Chapter IV., but also because if allowed to remain it would drop off after the paint was applied, and would then expose an unprotected surface to corrosion; and it is essential that all traces of rust should be removed, since iron in the presence of its own rust continues to be corroded even after the atmosphere is excluded by a covering of paint. Also, during the building of the vessel all the surfaces of the frames, plates and bars inside the ship, together with the side plating above the water line outside, receive three coats of the best paint for the same purpose.

The outside surface below water of iron and steel vessels has not only to be protected against the corrosive action of the sea water, but has to be also coated with some kind of *antifouling* composition which is intended to be gradually dissolved or washed off the vessel during her passage through the water, in order to carry with it adherent substances which would otherwise impede the ship's progress. Usually three coats of an anti-corrosive paint are first applied, and afterwards one of anti-fouling composition.

Fittings which are exposed to the weather, or to the condensation upon them of water vapour are *galvanized*; that is to say, after the article has first been placed for some time in a bath of hydrochloric (or *muriatic*) acid to clean it from grease and dirt-and then dried, it is lowered gradually into a bath of molten zinc; after a short time it is removed, and the layer of zinc attached to the article effectually prevents all corrosion.

The inner surfaces of the bottom plating are coated with cement on their lower parts, to prevent the mechanical wearing away of the rivet heads and plating through the constant wash of the bilge water and contained solid substances from side to side as the vessel rolls, and to prevent the corrosive action of the bilge water upon these parts, intensified as that action often is by the presence of acids (especially in merchant ships) due to the decomposition of coals, iron and copper pyrites, grain, etc., carried as cargo. The cement is thickest at the keel —extending up to the drainage holes cut in the frames—and thinner towards the bilges; the rivet heads are well covered, and all angles at the plate edges are filled up to prevent the lodgment of water. That this protective coating is necessary was demonstrated in the case of the store ship *Megaera*. This vessel, on her passage to Australia, sprang such a dangerous leak in her bottom plating under the bilge that she had to be run ashore. On inquiry, it was found that the part where the leak occurred had not been cemented, owing to its inaccessibility at the time, with the result that the plate had been completely eaten through. This points to the necessity for frequent inspection of these surfaces, and for the renewal of the cement where required.

Galvanic action occurs when two metals, having different positions in the electro-motive series, are placed in the same exciting liquid with a metallic connection between them, no matter how circuitous that connection may be; and it results in the dissolution of the electro-positive metal. The precautions taken to prevent this action between the steel hull and the copper of vessels sheathed with that material, have been already (page 107) pointed out; as has the necessity for removing the mill scale from all steel plates liable to immersion in sea or bilge water, such as bottom plates, the lower parts of frames and bulkheads, etc. The outside of an iron or steel vessel is liable to attack owing to the presence of metal propellers, Kingston and other valves, and of the metal gland round the rudder head where it passes through the stern post. The action of the first is lessened by coating them with the same compositions as the bottom plating, and is still further minimised by fitting a *zinc protector* to the hull, where the outer end of each propeller shaft is supported; the galvanic action resulting from the propellers then takes effect upon the zinc, the less electro-positive bottom plating remaining uninjured. Similar protectors are also attached to the hull near the other metal fittings just mentioned. It is to avoid the effects of galvanic action upon the bottoms of iron or steel ships where the paint has been worn off, that the Admiralty regulations order that such vessels are not to be secured to the same moorings as copper bottomed ships, nor to adjacent pairs of moorings attached to the same ground chains. To avoid injury to the inside of a ship, no copper or other metal fittings should rest directly upon the bottom, or be liable to immersion in bilge water; thus the ends of all copper, brass, and lead suction pipes are, where possible, made of zinc, or of zinced or enamelled iron. Where pipes of lead are unavoidably low in the ship, they are well coated with varnish to lessen their action upon adjacent parts of the vessel, and the paint and cement upon such parts should be specially well looked after and renewed as often as necessary; similarly placed copper or brass pipes are first varnished and then covered with canvas made waterproof by a coating of varnish. Care should also be exercised to see that no copper filings drop upon and rest in contact with the inside surfaces of the bottom plating.

CHAPTER XVI.

STRAINS EXPERIENCED BY SHIPS.

IN the preceeding pages structural details have been given for various classes of ships, and it now remains to point out the utility of those arrangements for resisting the many straining actions to which vessels are subject, both in still water and amongst waves.

By *strain* is meant the alteration, or tendency to alteration, of the shape of the vessel due to the operation of forces acting upon it. This alteration may take place in the structure *considered as a whole*, or in particular parts of it; in the first case the strains are said to be *structural*, and in the latter, *local.*

Structural strains may take place either in the longitudinal or transverse directions, and some of the principal causes producing these will now be referred to, observing that although box-shaped vessels have been taken as illustrations for the sake of simplicity, the conclusions reached are equally applicable to vessels of usual form.

In *still water*, longitudinal strains are brought about by an unequal longitudinal distribution of the weight and buoyancy. It will be remembered that for a ship to float at a certain water line its total weight must be equal to the buoyancy up to that line, and the centre of gravity must be in the same vertical line as the centre of buoyancy. Now, it is possible to place the component weights of the vessel in very different relative longitudinal positions whilst still satisfying the second condition mentioned above; but such differences of distribution greatly affect the longitudinal strains, as one or two simple examples will show. Suppose Fig. 127 to represent a vessel heavily laden with cargo, which

Fig 127.

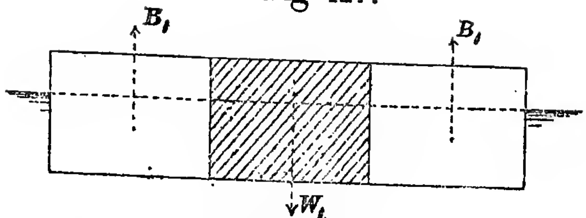
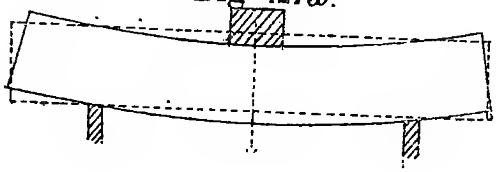


Fig 127a.



is concentrated over the middle third of the vessel's length, as shown. Regarding the ship as being made up of three equal lengths, each part is acted upon by one-third the total buoyancy ; but, on the end lengths, this is only opposed by the comparatively small weight of the framing, &c., of the vessel at those parts, and thus there is a large resultant upward pressure B_1 tons, say, on each end length. On the contrary, the total weight of the middle third of the vessel is greater than its buoyancy, and there is a resultant pressure, W_1 tons say, downwards, W_1 being equal to 2 B_1 because of the necessary equality between the total weight and buoyancy of the vessel. Such a ship is evidently in exactly the same position for being strained as the beam shown in Fig. 127*a*, which is supported near the ends and loaded in the middle ; under these circumstances the middle part of the vessel will tend to drop relatively to the ends (as in Fig. 127*a*), when she is said to *sag*, producing *compression* of the material, or a tendency to crush it up, at the upper part of the vessel, and an *extension* of the material, or a tendency to tear it, near the keel. At the same time, those parts of the vessel which are near the middle of its depth are only subjected to very moderate crushing or tearing stresses. Next suppose that the cargo of the vessel had been uniformly distributed over the whole length, as in Fig. 128 ; it is evident that the conditions of equilibrium before stated are satisfied in this as in the preceding case, but as the buoyancy of any length of the vessel is now equal to the total weight of that length, no strains result. Finally, if the cargo had been disposed as shown in Fig. 129, the vessel

Fig 128

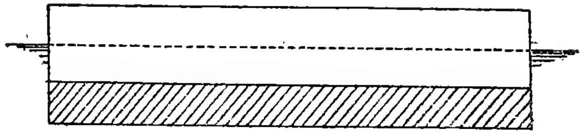


Fig 129

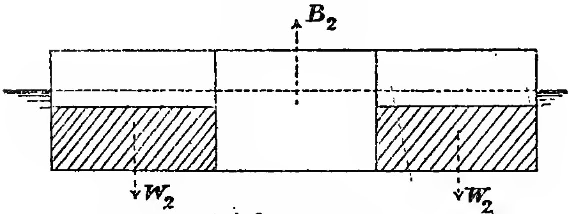
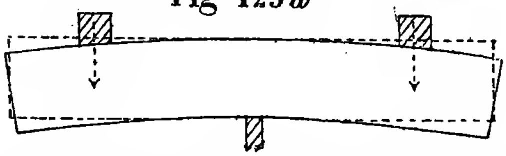


Fig 129a



would have been in a similar condition to the beam in Fig. 129*a*, which is supported in the middle and weighted at the ends. The ship would here tend to *hog*—the ends drooping relatively to the middle—and the parts near the keel would be subjected to compressive stresses, whilst the upper works would be in tension: as before, the parts near the middle of the depth would only be slightly affected. Thus it is seen that by altering the longitudinal distribution of the lading of this particular vessel, very different straining actions result, and this is also true of actual ships, differences in the placing of the component weights leading to differences in the strains experienced.

Amongst *waves* the longitudinal strains are much greater than in still water, and are constantly changing, whereas in the latter case they always remain the same for a given condition of stowage. Take, for example, the particular, simple, case of a box-shaped vessel meeting with waves of her own length, the cargo being supposed uniformly distributed longitudinally. At one instant

Fig 130.

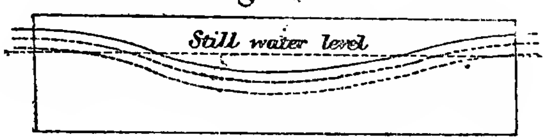
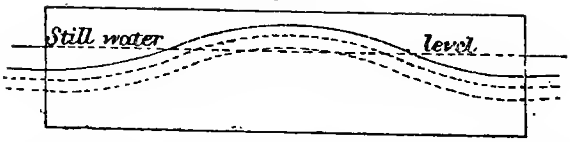


Fig 131



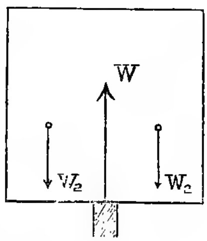
the buoyancy at each end will be augmented by the crest of a wave as in Fig. 130, whereas that amidships will be diminished by the occurrence there of a wave hollow. At such a time the vessel will therefore tend to *sag*. A few seconds later the wave will reach the position indicated in Fig. 131, a crest being in the middle and a hollow at each end, and *hogging* will result. Thus, as the waves pass the vessel, the varying distribution of the buoyancy produces forces which tend to bend the vessel first in one direction and then in another.

Severe longitudinal strains also result when a ship is left aground with a long part of its length unsupported, as, *e.g.*, a vessel across a sandbank in a tideway, having its ends left with little support as the tide falls.

In all cases of longitudinal bending the tendency is to break across a transverse section. A glance at the sections on Plates III. to V. will show how this tendency is resisted in the various ships; all parts which cross any section and have the several lengths of which they are composed joined together at their ends are available for resisting either tensile or compressive stresses at that section, whilst parts not so joined together can resist compressive stresses only. It is thus seen that the bottom and side plating (including the various sheer strakes) and planking is of immense use for this purpose, as is also the deck plating and planking; the parts of the former from the keel to the turn of the bilges, as also the plating and planking of the upper decks, are especially valuable, since it is at those parts that the maximum tensile and compressive stresses are produced. The *longitudinal* bulkheads and frames also contribute to the longitudinal strength, one of the principal functions of the latter in ships with double bottoms being to efficiently connect the two skins, so that they may act together in preventing the vessel from bending longitudinally; and it is with this object that the principal framing of such long and heavy vessels is placed in a fore and aft direction and made continuous. It will be noticed that as but small stresses have to be met in the middle of the section, it is not necessary to put much material there for structural strength, either at the sides of the vessel or on decks ; hence it follows that armoured sides and protective decks have their materials in by no means the best positions for contributing longitudinal strength to the structure, their disposition having been determined solely from considerations of the protection to be given against the attack of guns.

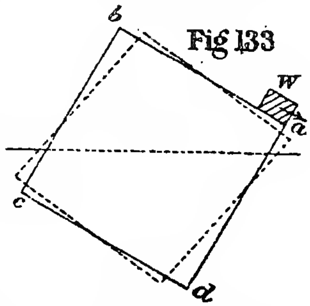
Coming now to the causes of transverse straining, one of these is the side pressure of the water, mentioned in Chapter I, such pressure tending to crush in the sides. A second cause operates when a vessel takes the ground. Suppose Fig. 132 represents a section of a ship which is resting upon her keel ; there will be an upward pressure at the keel equal to the weight W of the vessel, and a downward weight $W_2 = \frac{1}{2}W$ acting through the centre of gravity of each half of the ship as shown, such forces evidently tending to break the ship across near the keel. Similar straining occurs during the process of docking, though to a less extent, since shores are placed beneath the vessel as the water leaves her, which help to take her weight and lessen the strains.

Fig. 132.



Rolling motions are also influential in producing transverse alteration of form, and the general way in which they do so may perhaps be best understood by reference to a particular case. Let $a\ b\ c\ d$ in Fig. 133 be the position of a box-shaped vessel at the instant of reaching its maximum angle of roll, and suppose W is a heavy gun mounted on the broadside. When the vessel comes momentarily to rest, preparatory to returning towards the upright, the gun will tend to slip along the deck overboard ; if, however, it is well secured, a force is induced upon the deck in the direction indicated by the arrow, and this tends to distort the section into the form shown in dotted lines, so that the angles between the deck and sides are altered. When the ship reaches her maximum inclination on the opposite side, the section is racked as before, but in the contrary direction. What is true for this particular weight W holds good also for all the other weights in the ship, such as decks, etc., the general effect being to produce forces tending to alter the transverse form.

Fig 133



The various transverse straining forces are met by the transverse framing and bulkheads, in conjunction with the side and deck plating and planking. The bulkheads resist all alterations of form at the sections where they are fitted, if proper precautions are taken to stiffen them as described in Chapter X.; and the combination formed by each transverse rib with the associated beams and pillars also offers considerable resistance to change of transverse form, attached as this combination is to the bottom and deck plating and planking. To resist any tendency to alter the angles between the decks and the ship's sides, the ends of the deck beams are formed into *arms* as shown in the various sections before referred to.

The causes of *local* straining are very numerous, and only a few of the most important can be mentioned. Some of these have been already adverted to in preceding chapters, as, for example, the strains incidental to ramming and to the chafe of the cables, which are met by doubling the skin plating near the bow. Also those induced by the propellers in the parts near the stern, which are provided against by locally thickening the plating. In addition to these, strains occur at those parts of a ship where heavy weights are concentrated so that there is in consequence an excess of weight over buoyancy, as in Fig. 127, as this tends to make the ship at that part bulge downwards relatively to the remainder of the vessel. This is prevented by the longitudinal frames, etc., which pass continuously over that part, securing the portion having excess of weight to those parts where there is excess buoyancy. Also, these concentrated loads require specially strong supports in order to prevent local straining of the structure beneath them. Examples of such support will be found in the solid plate frames lightened with holes which are worked beneath the side armour of armour clads (*see* Fig. 73) and underneath the machinery spaces, and in the *box* and other specially strong beams (Fig. 70) fitted below turrets. Again, the movement of the engines brings forces into play upon the structure beneath, which are resisted in a ship having a double bottom by the lightened plate frames already mentioned, and in other vessels by specially stiffening the framing. The local strains brought about by a vessel taking the ground may next be alluded to, although little *special* provision is made in the ships of the Royal Navy to meet this rather remote contingency, reliance being placed upon the ordinary structural arrangements. With merchant vessels the case is very different, and since they frequently have to take the ground their transverse frames are more closely spaced than in war vessels of similar size.

The rounded bottom plating amidships is well adapted by its form to resist alteration of shape from water pressure, but the comparatively flat parts at the ends are likely to *pant* (*i.e.* move in and out) locally under it, which would cause leaky rivets. The plating is stiffened to resist this pressure by the transverse and longitudinal framing, and in the cruiser classes the closer spacing of the former as compared with that in an armoured vessel makes up in this respect for the fewer longitudinal frames employed. It will also be remembered that at the extremities of recent armoured vessels, the reduction of transverse strength due to the omission of an inner bottom is compensated for by a closer spacing of the transverse frames, and these also give good support to the outer bottom plating at the parts where it is specially liable to pant. In some vessels also, special longitudinal framing is introduced, to stiffen the plating at the ends to resist this water pressure.

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